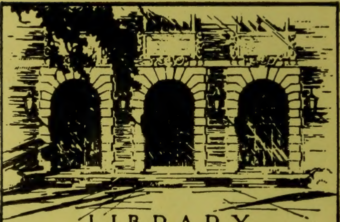


OPINIONS OF THE PRESS.

The Manufacturer and Builder, New York.

AN ENGINEER'S HANDY-BOOK.—Mr. Roper, the writer of this work, is well known to many of our readers as the author of a number of useful reference books relating to steam-engineering, which have become deservedly popular by reason of their plain and intelligible style and their freedom from unnecessary and confusing mathematical technicalities. Mr. Roper's object in all these hand-books has avowedly been to present facts and explain principles in language so plain and comprehensible that average steam-users, engineers, firemen, and those who are usually found in charge of steam-machinery, can read his books understandingly and with profit. We would be glad to see Roper's hand-books largely multiplied and distributed in every workshop, for it is only out of books of this kind that the average workman will be able to master the principles of his handiwork. The present volume is no exception to this rule; on the contrary, we regard it to be decidedly the best of Mr. Roper's books, both with regard to its substance and the manner in which the same is classified and presented.

The Locomotive, Hartford, Conn.

ROPER'S ENGINEER'S HANDY-BOOK.—Published by E. Claxton & Co., of Philadelphia, who are the publishers of several works on steam, steam-boilers, and engines, from Mr. Roper's pen. This last work is of special value to all who have to do with steam-boilers and engines, and it will be found a valuable shop companion for the mechanic. There are a great many facts collated that are not easily reached except through expensive books and libraries. These will be found of service to all classes of men, whether in trade or manufacturing. We commend it heartily, and believe it will have a large sale.

National Car-Builder, New York.

ROPER'S ENGINEER'S HANDY-BOOK.—This compact and comprehensive little volume contains a vast amount of information relative to the care and management of every class of steam-engines. It is profusely illustrated, and abounds in facts, figures, rules, tables, questions and answers, formulæ, etc., that are exceedingly valuable to engineers, and of easy reference by means of a copious and well-arranged index. The various subjects are discussed with brevity and

clearness, and with a freedom from technicality which enables the reader to get at the pith of the matter without fishing it out from an ocean of words. A prominent feature of the book is a full explanation of the steam-engine indicator, and its use and advantages to engineers and others. The long experience of the author in this branch of engineering, and the numerous publications he has already issued upon kindred subjects, give an additional value to the present treatise. It is printed on thin paper and in clear type, and contains 678 pages. Flexible tuck binding, gilt edge, suitable for the pocket.

Forest, Forge, and Farm, Ilion, New York.

ENGINEER'S HANDY-BOOK.— We have received a book with the above title, by the well-known author and engineer, Stephen Roper, who has written a number of works on the subject of engineering. The eminent reputation of the author is a sufficient guarantee that the book is both interesting and useful. Mr. Roper has had an experience of over thirty-five years with all kinds of engines and boilers, and thoroughly understands locomotive, fire, marine, and stationary engines. This work has 678 pages, is profusely illustrated, bound in morocco, and contains nearly 300 main subjects, 1316 paragraphs, 876 questions and answers, 52 suggestions and instructions, 105 rules, formulæ, and examples, 149 tables, 195 illustrations, 31 indicator diagrams, and 167 technical terms; over 3000 different subjects, with the questions most likely to be asked when under examination, before being commissioned as an engineer in the U. S. Navy or revenue service; before being licensed as an engineer in the mercantile marine service, or receiving a certificate to take charge of a steam-engine or boiler in locations where such certificate is necessary.

It is a very valuable book for engineers, and will no doubt meet with a ready sale. E. Claxton & Co., Philadelphia, are the publishers.

Leffel's Illustrated News, Springfield, Ohio.

ENGINEER'S HANDY-BOOK: By Stephen Roper, Engineer.— The author of the valuable series of hand-books which we have before referred to, has just issued the above-named work, which must find ready way into the hands of engineers and steam-users throughout the entire land. It contains a full explanation of the steam-engine indicator, its uses and advantages, with formulæ for estimating the power of all classes of steam-engines; also facts, figures, questions and tables for engineers who wish to qualify themselves for the United States navy, the revenue service, the merchant marine, or the better class of stationary engines. The work does not claim to teach how to design or proportion steam-engines and boilers, but rather to inform the engineer how to manage them intelligently. It is one of the kind of practical hand-books for which there is always need. The work is well bound in flexible leather, uniform with Roper's other hand-books, has 678 pages, and is fully illustrated.

American Machinist, New York.

Roper's Engineer's Handy-Book.— The subjects in this work have been treated in a brief and comprehensive way, therefore the reader is not required to read a number of chapters in order to acquire a little knowledge. The use of the indicator is treated in a plain, practical way, so that it may be readily understood. Abstruse formulas have been omitted and simple arithmetic used, thus avoiding the usual vexations among practical men, who are uneducated in the higher mathematics. The author has in this book given the results of his own practical experience, which extends over a period of thirty years and upwards, and the work will doubtless be read with pleasure and profit by very many practical mechanics.

Boston Journal of Commerce.

Mr. Stephen Roper is well known as the author of several other handy-books that treat on steam, steam-boilers, and engines. This new work is, in our judgment, his best. Although the arrangement and classification seem a little peculiar, and a decided new departure in book-making, they do not detract from the merit of the book, which is plain, comprehensive, and instructive from the title-page to "The End." It is neatly illustrated, and creditable in the highest degree to both author and publishers. It will be a valuable addition to every engineer's library.

Millstone, Indianapolis, Ind.

"The Engineer's Handy-Book," by Stephen Roper, Engineer, is a practical treatise on the management of the steam-engine. The author says the book was "not written for the purpose of instructing engineers how to design or proportion steam-engines or boilers, but rather to inform them how to take care of and manage them intelligently." The declaration is carried out in the plainest and most systematic manner. There is no straining after possibilities, but the facts, as a thorough mechanic and engineer understands them, are set forth in positive language and plain terms. This gives value to the work as a hand-book to such engineers who are not too egotistical to receive information. As a text-book for students in mechanical engineering, it will be found of great value. Its illustrations and tabulated matter are important features, and printed in the excellent style that characterizes all the books issued from the house of E. Claxton & Co., Philadelphia. It is something that should be possessed by every engineer.

The American Engineer, Chicago, Ill.

The Engineer's Handy-Book.—We are in receipt of the above work, which contains a description of the various forms of engines now in use, and supplies interesting and useful information as to the care, management, and remedy of

defects of steam machinery and its appendages, with tables for calculating the power of engines. Mr. Roper in his preface says: "This book was not written for the purpose of instructing engineers how to design or proportion steam-engines or boilers, but rather to inform them how to take care of and manage them intelligently, as well as to furnish to those intending to qualify themselves for the United States Navy, Revenue Service, Mercantile Marine, or to take charge of the better class of stationary steam-engines, with a plain, practical treatise." It is from this standpoint, therefore, that the book ought to be judged, and we are sure that the large class to whom it is especially addressed will find it a useful appendage and book of reference in their daily work.

The Scientific American, New York.

A well-made pocket-book of practical information for mechanical engineers, particularly those of limited education, and such as may wish to qualify themselves for service in the U. S. Navy or the mercantile marine. The more important engines in use are clearly described, and formulæ are given for estimating their power. Particular attention is paid to the Steam-Engine Indicator, its use and advantages. The author has had much experience in this class of work, and writes clearly and plainly.

Engineering News, New York.

An "ENGINEER'S HANDY-BOOK."—As a writer on subjects relating to steam and steam engineering, Mr. Roper is now too well known to need any further introduction. In this, his latest contribution to steam-engineering literature, Mr. Roper has aimed to present to his brother engineers a "handy-book" that will be to them what Trautwine's "Pocket-Book" is to civil engineers, and in doing this he has spared no labor in collecting and editing his materials. Some idea of the completeness of the work may be gathered from the statement of the publishers that it contains nearly 300 main subjects, 1316 paragraphs, 876 questions and answers, 52 suggestions and instructions, 105 rules, formulæ, and examples, 149 tables, 164 illustrations, 31 indicator diagrams, and 167 technical terms; over 3,000 different subjects, with the questions most likely to be asked when under examination, before being commissioned as an engineer in the U. S. Navy or Revenue Service; before being licensed as an engineer in the Mercantile Marine Service, or receiving a certificate to take charge of a steam-engine or boiler in locations where such certificate is necessary. The author does not claim to have discovered any recent special facts relating to his subject, neither does he claim to have written a book of instructions in designing or proportioning steam-engines; he aims rather to instruct how to care for and manage them intelligently. The book is very full and complete, and its typographical execution is perfect. It must readily recommend itself as an "ever-ready companion" to every steam-engineer in the country.

From the Textile Colorist, Philadelphia.

THE ENGINEER'S HANDY-BOOK.—Another aid to engineering by a well known author, who has already done much in the way of practically educating scientific students. The work before us is one of 678 pages of the most useful information. It treats exhaustively on the most recently invented adjuncts to the steam-engine, and gives very full formula by which engineers can accurately calculate power and make reliable estimates in all branches of their profession. It likewise presents the most desirable instructions to young men wishing to stand examination for the United States Navy or Revenue Service, as well as the merchant marine. It is fully illustrated, and got up in a style commendable in the publishers and flattering to the author.

From the American Manufacturer and Iron World, Pittsburg, Pa.

THE ENGINEER'S HANDY-BOOK: By Stephen Roper, Engineer.—Mr. Roper's name is by no means unfamiliar to the readers of popular steam-engineering literature in this country. The book now under notice is his last, and we believe the largest in bulk and the most comprehensive in scope of any work yet published by him. He has gathered into this single volume about all the practical information relating to the care and management of a steam-engine that one employed as a steam-engineer would be likely to require in ordinary service. The leading steam-engines now in the market are illustrated in this book with more or less descriptive matter accompanying each, giving the reader a general idea of the design of the engines, the details of their construction and operation. The various accessories to engines and boilers receive their full share of attention. The chapter relating to the indicator is well illustrated by means of numerous and well chosen diagrams.

The paper, press-work, and the general make up of the book leave nothing to be desired. The style in which the book is written will commend itself to those who want a book to read, and, thererore, free from mathematical formulæ and we have no doubt that the class of persons whom Mr. Roper addresses will find in this book all they will be likely to want in connection with any question relating to the steam-engine.

OPINIONS OF EMINENT ENGINEERS, ETC.

THE FOLLOWING LETTERS HAVE BEEN RECEIVED FROM SOME OF THE MOST DISTINGUISHED MECHANICAL ENGINEERS, EXPERTS, AND AUTHORS IN THE COUNTRY.

E. Claxton & Co. *Cincinnati, Ohio, Aug. 3, 1880.*

Permit me to acknowledge copy of your Roper's Engineer's Handy-Book. The volume contains a large amount of useful information for students of mechanical engineering, arranged in a condensed form, and cannot fail to be a valuable acquisition to young engineers and mechanics. *JOHN W. HILL.*

E. Claxton & Co. *Yonkers, N. Y., Aug. 14, 1880.*

In reply to your favor of the 2d instant, I would say that I think Mr. Roper's "Engineer's Handy-Book" is the best one of his recent works. Permit me to say that, when asked, as I often am, by the men I meet in charge of engines, as to what books they had best get "to read up on the engine," I say "get Roper's Works." In the future, as in the past, I shall take pleasure in endorsing his effort to the men for whom he has written. *W. H. ODELL.*

Messrs. E. Claxton & Co. *Troy, N. Y., Aug. 11, 1880.*

Your favor of a late date, as well as enclosure of "Roper's Engineer's Handy-Book," duly received. Permit me to say, that I think the book a very valuable addition to the literature of the subjects of which it treats; and, while the accomplished engineer will find in this book many facts so plainly stated as to save much time in working up, the intelligent engineer and mechanic, whose opportunities in the past have hardly permitted his becoming fitted, and whose time in the present will hardly allow him to wade through the verbiage and mathematical demonstrations in which such knowledge as is contained in Mr. Roper's book is usually enveloped, will find in it a large amount of information stated in the common language of every-day life. Such books cannot be too widely distributed. The time was *when their possession was a convenience. The time* is *when their possession is almost, if not quite, a necessity.* *F. F. HEMENWAY.*

E. Claxton & Co. *Passaic, N. J., Aug. 14, 1880.*

I have examined "Roper's Handy-Book" pretty thoroughly, and have no hesitancy in pronouncing the work an excellent one. It is decidedly out of the beaten track, and the better for it.

WM. H. HOFFMAN.

E. Claxton & Co. *Chicago, Ill., Aug. 18, 1880.*

Your note of July 29th, and a copy of "Roper's Engineer's Handy-Book," were duly received. The book is well calculated to accomplish the purpose of the author, viz., to furnish practical and valuable information to engineers. The comprehensive description of the various types of automatic engines is a fund of useful knowledge, and the various groups of questions, the answers to which are embodied in the text, are very likely to cause readers to "think," and to fasten the ideas in their minds. The book is a desirable addition to an engineer's library.

CHARLES A. HAGUE.

E. Claxton & Co. *Hamilton, Ohio, Aug. 30, 1880.*

Your esteemed favor, and also a copy of Roper's Engineer's Handy-Book, were duly received; and, in reply, I beg leave to say that the work is well got up, and I consider it of more practical value than any I have yet seen. It seems almost impossible to find steam literature adapted to the wants of steam users. This book fulfils this requirement, and deserves a good reception at the hands of a class of men whom it may greatly benefit. *J. W. SEE.*

E. Claxton & Co. *Hartford, Conn., Sept. 8, 1880.*

Your favor of the 29th of July came in my absence. I have just returned, and hasten to reply. Roper's Engineer's Handy-Book is replete with the information that Engineers need at hand. It combines such portions of more pretentious works not readily accessible to the Engineer, as well as information from Mr. Roper's wide practical experience with the detailed working of Boilers and Engines, as will give it value in the Shop and Engine-room. But it has a wider range than this. It contains valuable tables, articles on the U. S. Naval Service, Revenue Service, and Mercantile Service, with qualifications required of persons seeking appointments in each, and numerous other matters that make the work a very valuable compendium. I shall keep a copy on my desk ready for reference, and cheerfully commend it to others. Business men and manufacturers will find it a very convenient Hand-Book. *J. M. ALLEN,*

President Hartford Steam-Boiler Inspection and Insurance Company.

E. Claxton & Co. *Columbus, Ohio, Oct. 23, 1880.*

I esteem this book highly as one containing much information not found elsewhere in a condensed form. It bears directly on practical questions in mechanical engineering, especially on matters pertaining to the Indicator and its use. How to read a diagram and determine the condition of the action of engines are made clear.

The book is of especial value to any who may be interested in the peculiarities of existing engines, as I find the book contains illustrations and descriptions of most of the prominent engines in use.

S. W. ROBINSON,

Prof. of Mechanical Engineering, Ohio State University.

E. Claxton & Co. *Boston, Nov. 24, 1880.*

I can give my opinion of Roper's Engineer's Handy-Book in a few words: it is the book that has been needed for more than 50 years. It is the only book on steam and the steam-engine that I know of which is devoid of the mysteries of algebraical formulæ, and which the engineer or student, with only a common-school education, can read and understand; it consequently leaves no excuse for the ordinary engineer to be ignorant of the principles of steam-engines.

F. W. BACON, M.E.

E. Claxton & Co. *Hoboken, N. J., Dec. 31, 1880.*

Gentlemen:—I am in receipt of your favor and also a copy of Roper's Engineer's Handy-Book; please accept thanks for the same. I am too much occupied with collegerwork at the present time to give it a complete analysis, but at a cursory glance I see it is full of valuable information for those who use or handle steam-engines, and should think it would have a very extensive sale.

R. H. THURSTON,

Prof. of Mechanical Science, Stevens Institute of Technology.

ENGINEER'S HANDY-BOOK.

CONTAINING

A FULL EXPLANATION OF THE STEAM-ENGINE INDICATOR, AND ITS USE AND ADVANTAGES TO ENGINEERS AND STEAM USERS.

WITH FORMULÆ

FOR ESTIMATING THE POWER OF ALL CLASSES OF STEAM-ENGINES; ALSO, FACTS, FIGURES, QUESTIONS, AND TABLES FOR ENGINEERS WHO WISH TO QUALIFY THEMSELVES FOR THE UNITED STATES NAVY, THE REVENUE SERVICE, THE MERCANTILE MARINE, OR TO TAKE CHARGE OF THE BETTER CLASS OF STATIONARY STEAM-ENGINES.

With Illustrations.

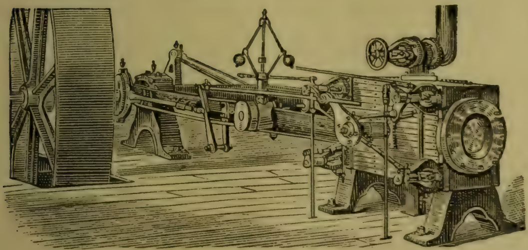
BY

STEPHEN ROPER, ENGINEER,

Author of

"Roper's Catechism of High-Pressure or Non-Condensing Steam-Engines," "Roper's Hand-Book of the Locomotive," "Roper's Hand-Book of Land and Marine Engines," "Roper's Hand-Book of Modern Steam-Fire Engines," "Improvements in Steam-Engines," "Use and Abuse of the Steam-Boiler," "Questions and Answers for Engineers," etc., etc.

THIRTEENTH EDITION.



PHILADELPHIA:

EDWARD MEEKS,

1012 WALNUT STREET.

1892.

INTRODUCTION.

IT is quite customary for persons to write books on the steam-engine, and then offer as an apology for so doing, that they have discovered that there is no practical treatise on the same subject in the market, which shows either a lack of modesty on their part, and a want of appreciation of what has already been written, or an unwillingness to do justice to those who have previously treated the same subject. There is no want of literature on the steam-engine; in fact, it would be difficult for the most experienced engineer or talented author to add anything original. The steam-engine of the present day is probably as perfect as it ever will be; in fact, there has not been any important improvement made in any class of steam-engines for several years, except in the quality of the materials employed in their construction and refinement of workmanship; consequently, the work of those who treat on the steam-engine, for the present, must be confined simply to abbreviating, simplifying, correcting, and explaining what has already been written, as well as noting the results of the experiments which are tried to test the efficiency of different designs of steam-engines. Whoever will apply himself to this object in the future, will be performing what has long been needed. Of course, we may discover a new engine that will be radically different from any in use at the present day, which would involve the necessity of a new order of literature and new theories, but such an innovation is highly improbable, and casts only a dim shadow in the future.

This book was not written for the purpose of instructing engineers how to design or proportion steam-engines or boilers, but rather to inform them how to take care of and manage them intelligently.

as well as to furnish to those intending to qualify themselves for the United States Navy, Revenue Service, Mercantile Marine, or to take charge of the better class of stationary steam-engines, with a plain, practical treatise. In order to enhance its value to young engineers, as well as those of limited education, none but the plainest language has been used. This has not been done for the purpose of encouraging the engineer to dispense with the use of mathematics, or discard theories, as all our great triumphs in mechanical science have been based on theories and demonstrated by practice.

In the discussion of the different subjects brevity has been adhered to, because the spirit of the age demands it, even in the discussion of the most important subjects. There can be no reason why the reader should be compelled to wade through chapters of matter to obtain information which may be condensed into a few terse and intelligent paragraphs, nor to deal with the dead past when the living present is before him. The mathematical formulæ employed have been abbreviated, since it is immaterial how a problem is worked, providing the result is correct and susceptible of easy explanation. Up to the present time, the knowledge to intelligently apply the steam-engine indicator has been confined to a few persons in every country styling themselves experts. This partly arose from the fact that authors who have heretofore treated on this subject were men of literary ability and well versed in mathematics, who found it more agreeable to elucidate their subject in their own peculiar style than in any other.

The writer's experience of over thirty-five years, and his association with all classes of engineers, enable him to understand fully the kind of information most needed by the average engineer. Consequently, he has undertaken the task of furnishing it, and how well he has succeeded in the accomplishment of his object, he cheerfully leaves to the reader to decide. If it should appear that he has succeeded in imparting useful and important information to the members of a profession to which he himself belongs, he will feel amply rewarded for his efforts.

S. R.

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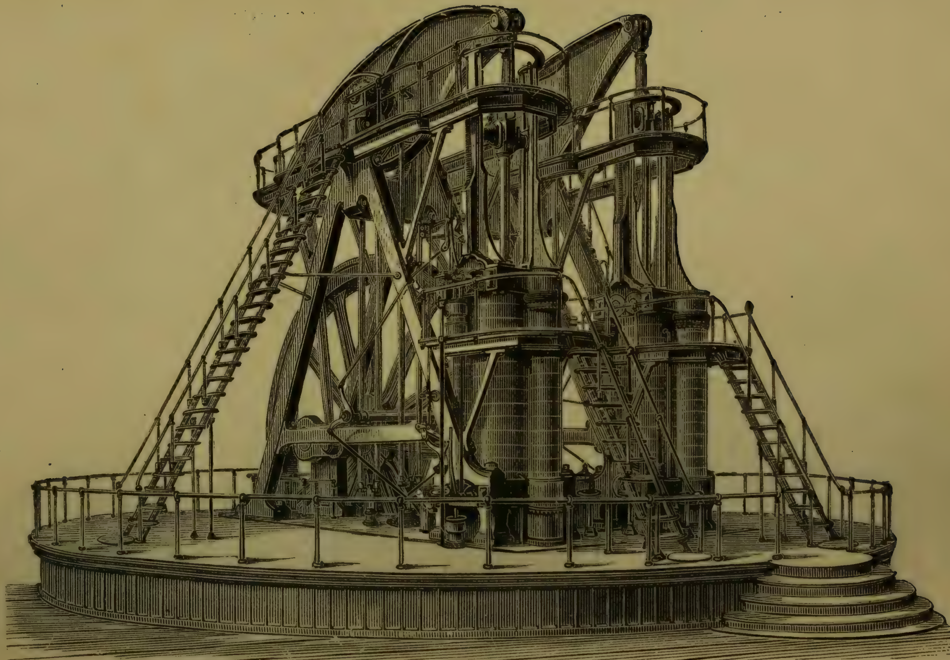


The Centennial Corliss Engine.

THE

ENGINEER'S HANDY-BOOK

PART FIRST.

The Centennial Corliss Engines.

The Centennial Corliss Engines were beam-engines of the Corliss type, with all the latest improvements, and nominally of 700 horse-power each, or 1400 horse-power together. The cylinders were 40 inches in diameter, with 10 feet stroke. They were provided with air-pumps and condensers, consequently they could be worked either condensing or non-condensing, and were intended to work with from 25 to 80 lbs. of steam pressure, according to the requirements of the exhibition. The gear fly-wheel was 30 feet in diameter, 2 foot face, and weighed 56 tons; it was undoubtedly the largest gear-wheel ever made. The pinion that was driven by this large fly-wheel was a solid casting 10 feet in diameter, weighing 17,000 lbs., and was the largest ever made. The main frame was A shaped, having the journals for the beam-centres on the top, and the legs bolted to the bottom of the cylinder on one side, and to the main crank-shaft journals on the other.

The walking-beams were of the web-beam pattern, and made of cast-iron, and, in consequence of their peculiar shape, detracted very much from the general appearance of the engine. They were 9 feet wide at the centre, and 27 feet long, each weighing 22,000 lbs. The cross-head guides extended from the upper cylinder-heads to the top gallery, and were provided with screws by which they, as well as the cylinder-heads, might be lifted to admit of access to the pistons. The piston-rods, which were made of steel, were $6\frac{1}{4}$ inches in diameter. The cranks were highly finished, and weighed 10,000 lbs. each. The connecting-rods were 24 feet long. The steam-valves received their motion from a wrist-plate and a system of levers similar to those employed in the ordinary Corliss engine, and the releasing gear for them was entirely original, and very ingenious, though the exhaust-valves ended their vibrations by an abrupt kick or jerk.

The height of the engines, from the floor to the top of the walking-beams, was 39 feet, and their weight, with all their adjuncts and attachments, was over 700 tons. The engines were supplied with steam by 20 upright Corliss boilers, of 70 horse-power each. The main steam-pipe was 18 inches in diameter and 320 feet long. The engines rested on a platform 55 feet in diameter, and $3\frac{1}{2}$ feet above the floor of the building. The top of the frame was surrounded by a circular gallery, which afforded access to the beams and all the upper works; this gallery was reached by a semi-circular stairway on each side. These engines were objects of general interest and curiosity, and served to illustrate the wonderful development of the steam-engine in this country, and the amount of inventive genius that must have been devoted to its improvement.

After the close of the Centennial they were taken down and removed to the builder's establishment, Providence, R. I., where they remained until recently, when they were sold to the Pullman Palace Car Co. for the purpose of furnishing the motive-power for their works, near Chicago, and also the power for the Allen Paper Car-Wheel Works adjoining.

Steam Engineering.

Steam Engineering has assumed such vast proportions as an agent of modern progress and civilization, that it has given birth to a profession whose scope and functions are not yet very clearly defined. The engineer's duty, in the performance of his daily routine, involves the application of the laws of Nature in various ways, to understand and explain which require a wide range of scientific knowledge. While there are to be found in the profession men whose intelligence and acquirements would shed lustre on any calling, there are others who, by their loose disregard of correct rules, show that they are laggard in the acquisition of that real knowledge so essential to men in their profession. This is to be regretted, in view of the vast amount of property and the great number of valuable lives intrusted to their care, both on sea and land. But whenever any attempt is made to induce engineers to qualify themselves for their calling, the effort is met with the old stereotyped question regarding the relative merits of theoretical and practical engineers, or the comparative value of theory and practice. The practical men, who have no theoretical knowledge, scoff at the theorists, and the latter sneer at the former. It requires very little experience on the one hand, and not much study on the other, to show that each are equally important, only in different ways. Both parties should know that "Theory and Practice make perfect." Theory, together with practical experience, will, without doubt, enable men to excel in whatever work they may undertake. Therefore, it should be the highest ambition of engineers to combine theory with practice, and prove the one by the other.

This object may be effected by devoting a portion of their leisure hours to study, and by pursuing a systematic course of self-culture. The engineer whose early training has been neglected, and who is now debarred from the advantages of a good education, need have no cause for despondency, because the extra exertion and effort required to educate himself will confer advantages

of their own, which the routine work of a school cannot develop. Of course, there are men in this, as in all other callings, who will fail, however much they may try to accomplish in the way of educating themselves. This arises from the fact that, though morally all men may be equal, intellectually they never can be so. Consequently, the ability of men to educate themselves varies in proportion to the amount of natural intelligence they possess. But in any case, study gives quickness of apprehension, enables a man to profit by all the recorded experience of others, develops a power of appreciation and concentration, enforces exactness and accuracy, and, if properly directed, teaches men to classify facts, make proper deductions and reason logically. The knowledge acquired from the study of books is of inestimable value to the young engineer, as without it he can never be thoroughly qualified for the duties of his profession, since he will be lacking in certain definite information which can only be obtained from them, owing to the want of which he is almost sure to be not only narrow-minded, but also very slow to receive new ideas or to estimate the proper value of old ones.

Such persons, if occupying positions in which they exercise authority, are very apt to become intolerant of other people's opinions, to assume that all knowledge begins and ends with themselves, or with what they have learned, and to over-estimate their own ability. They are apt to be self-conceited, a quality which too many in every calling possess, mistaking it for an independent spirit. One of the commonest excuses for ignorance is the stereotyped expression, "I am too old to learn." This, if made in sincerity, is a great mistake, as it is a false pride which neglects an opportunity to learn because it comes late in life, and it is a false fear which shrinks from an effort on account of its difficulty. One fact very important to be considered in this connection, is, that knowledge throws light upon itself; and that it is the first step only that must be taken gropingly, as it were in the dark, as the bugbears in such cases, like shadows, vanish the moment they are boldly approached, and will be found to be mere shadows after

all. Truths are in the main simple and easy to be understood, and are daily being brought more within the grasp of the most ordinary comprehension by means of good books, which may be had at trifling cost. It is frequently asserted, by members of this calling, that they are no book-engineers; which statement betrays their ignorance of the manner in which some of the most valuable books on the steam-engine originated. They were written by engineers of experience, who wished to advance their profession, and who thought that, if their predecessors could commence their studies in their young days, they themselves might advance and improve still further, leaving the benefit of their experience to posterity; the art would therefore advance with the age. As much information may be learned in a few weeks from the works which they have left us, as had taken them years of observation and trial to ascertain.

Most of the abuses connected with steam engineering have arisen from two causes, viz., avarice and ignorance; avarice on the part of owners of steam-engines and steam-boilers, who entertain the idea that cheap steam-engines and boilers might be managed by a class of persons who were willing to work for very low wages; and ignorance on the part of those who claimed to be engineers, but who were only men of all work, or at best mere laborers in the treadmill of routine (stoppers and starters). It is evidently one of the greatest mistakes connected with the use of machinery, to intrust its care and management to persons of inferior judgment, as a competent engineer, who could command good wages, would probably save three times the difference by his judgment and skill in its proper maintenance. If engineers wish to raise the standard of their profession to what it ought to be, and command remunerative compensation for their services, they may do so by educating themselves, and not otherwise. It will not do for them to shrug their shoulders, and claim to be "practical men," who reject theories, because it is well known that such men have become a nuisance in every branch of mechanics, being the least progressive, the least enlightened, and the most stubborn

in the assertion of their views; because their minds are cramped, and will not allow of either the substitution or the admission of ideas different from their own, however crude and primitive they may be.

The engineer of the ferry-boat *Westfield* belonged to this class. Although he had been fourteen years an engineer on tug- and ferry-boats, he was unable to tell the figures on the steam-gauge; and, at the investigation that followed the frightful disaster that occurred on board that ill-fated boat, on being asked what a vacuum was, answered that "he thought it was foul air." It was also in evidence that the chief engineer of the line on which he was employed was equally wanting in that practical knowledge that ought to be possessed by a person occupying his position. These may, perhaps, be said to be extreme cases, but they will only prove to be so when it can be shown that there are not hundreds of others occupying the same positions who are not much better informed. No man is practical unless he proves practice by theory and theory by practice, and who attaches any importance to statements not sustained by facts. Such men can always be distinguished from the self-styled "practical men," by an unassuming manner, and by rarely making any pretensions; when expressing their opinions, they have a tendency to underrate their own ability, not because they pretend to be less capable than they really are, but (as so many men have become pretentious in their manners and expressions) because they fear they may be considered as belonging to that class. On the other hand, the self-styled variety are continually thrusting themselves forward, and can easily be distinguished by the profuse use of the pronoun "I," which is evidence of conceit or ignorance, or perhaps of both.

A great deal has been said and written on the subject of licensing engineers, but there seems still to be as great a diversity of opinion, as to the benefit to be derived from it, as on any other connected with the profession. Many engineers are of the opinion that, in consequence of the loose and uncertain way in which examinations for licenses are now conducted, a law that would

require every engineer to submit to a rigid examination, would prevent all but first-class men from being employed as engineers. But while all agree that there should be a license law to reach all classes of engineers, there are more formidable difficulties to be overcome in the impartial execution of such a law than appear at first sight. In the first place, it would be almost impossible to place the office of examiner or inspector beyond the reach of political influence, consequently his decisions would, in many instances, be likely to be influenced by partisan feelings. The next objection is, that it is not in the power of any man to determine, with any certainty, the ability of an engineer by any theoretical examinations. The candidate should be required to show his ability by practical demonstration. Another point is, that it is very difficult for a stranger to judge a man's qualifications as an engineer, with any degree of certainty, in comparison with those who are in daily intercourse with him, which goes to show that, unless it is possible to determine to a certainty a man's ability as an engineer, the license is of no value. The execution of any license law to produce beneficial and satisfactory results should only be intrusted to a board of engineers, composed of theoretical, practical, and painstaking men — men who have performed all the duties incidental to the calling of an engineer.

For this reason examinations ought to be conducted in the engine- and boiler-rooms, where the persons applying for certificates are employed. In that case, there would be an opportunity to test the candidate's practical knowledge of everything connected with the engine and boilers under his charge. There can be no reason why persons, whose duty it is to inquire into the capabilities of persons having charge of steam-engines and boilers, should not do so on the premises, or on the vessels on which they are employed, as well as to have them go several miles, and frequently into another county, for that purpose. Examinations ought to be uniform in all localities; as, where the subjects embraced in the examination differ in different localities, the system is unjust.

There are thousands of instances on record where men, having charge of engines and boilers for 10 or 12 years, have secured only a second- or third-class certificate, simply because they were men of limited education, and could only imperfectly express what they actually knew; while others, who could furnish no positive evidence of ever having had charge of an engine or boiler, and who did not possess any of the qualifications so essential to an engineer, obtained first-class certificates, because they were theorists and good mathematicians. It is quite common to find blatant individuals, who have no reputation for ability, sobriety, and industry, parading first-class certificates, which they obtained because they had abundance of assurance, while many practical and unassuming men are almost afraid to apply for a certificate, lest they should be degraded to the level of a third or fourth class engineer. While theorists and mathematicians should receive their due meed of merit, it would seem unjust, so far as the awarding certificates is concerned, to place them above the men who, though possessing only a limited education, had shown by years of industry, truthfulness, and the successful pursuit of their calling, that they were perfectly reliable in every respect. These are nice points to decide, particularly when it has to be done by one man, perhaps without any practical experience.

The character of steam engineering can never be much elevated by examinations and the awarding of certificates; the only hope for this lies in a law requiring every man to possess an elementary knowledge of steam and steam machinery before being permitted to take charge of an engine and boiler; as, being once recognized as engineers, however ignorant men may be, they, as a general thing, evince a lack of interest in acquiring a more extended knowledge of the duties of their calling. They frequently become too conceited to take instructions from others, or even to ask a question, although the answer might put them in possession of a fact of immense value to them. There is no reason why one class of men should be required to serve a regular apprenticeship, and even to devote years to the study of their profession, while

another class is allowed to discharge all the duties of a calling equally as important, with scarcely any preparation.

The question is often asked, "Should an engineer be a machinist?" The proper answer would be, not necessarily so; there is no reason why a man should learn two trades in order to follow one. Besides, experience has shown that, though machinists are in some instances the best judges of things that may transpire in relation to steam machinery, they are, nevertheless, frequently less careful, less reliable, and less ingenious, than those who never learned a regular trade. Moreover, neither Savary, Smeaton, Watt, Stephenson, Fitch, Fulton, or either of the Stevenses, Baldwin, or Oliver Evans were machinists. An engineer should be possessed of natural talent, should be ingenious and able to discover any defect that may occur in the machinery under his charge, be able to take up the lost motion, or to take apart and put together the different parts of an engine.

There is great need of reform in the use of the term engineer, as a customary neglect to designate to what branch of the calling he may belong gives rise to much inconvenience and confusion. A bookseller advertises a book entitled "Hints to Young Engineers." Many men having charge of steam-engines order the book by mail, under the impression that it contains useful, if not valuable, information regarding their trade. On examination it may be found to be a treatise adapted only to young men preparing themselves for the calling of civil engineers, not making a single allusion, or having any bearing whatever, on the business in which the party ordering it was engaged. Another author writes a book, and terms it "an Engineer's Pocket-Book," intending it, of course, to be a hand-book for all classes of engineers, as in the former case; although it may contain a good deal of valuable and useful information, it will be found, nevertheless, too limited to meet the requirements of any one class, as it would be impossible to embody such information in a book of ordinary size, or in any book that would come within the reach of persons of limited means; nor would it be possible for any author, however

learned he may be, to elaborate so great a variety of subjects, requiring, as they would, scientific accuracy and mathematical precision.

It is not uncommon to find men who have been educated as civil engineers, and who have devoted their lives to the pursuit of that calling, presuming to write treatises for the instruction of mechanical engineers, or men having charge of steam-engines and steam machinery, without possessing the first qualifications for such an undertaking. With the same propriety, the lawyer might write a treatise for the instruction of the doctor, and *vice versâ;* or the doctor might attach the word squire to his name, and the lawyer appropriate the title of M.D. It would be more appropriate to use the terms, mechanical or steam engineer, civil engineer, hydraulic engineer, dynamic engineer, sanitary engineer, etc., as, by the general adoption of these terms, persons wishing information in regard to machinery, bridges, embankments, or hydraulics, might consult the right person, instead of being subjected to the annoyance that is frequently experienced in consequence of consulting or engaging the services of the wrong party. Engineers of every class are very useful, though in different ways; their labors, next to that rational intellect which places man above the beast, have conferred on mankind the greatest boons, and the monuments which display the conceptions of their genius are almost as indestructible as the firmament or the ocean. It cannot be said of the engineer, as has been frequently said of the lawyer or the doctor, that if mankind could do without him it would be well for the human race.

Facts that should be Borne in Mind by Engineers.

No man who loves exact knowledge can fail to find scope for the exercise of his intellect in the calling of an engineer, as it is adapted to men of the most opposite temperaments. Two conditions alone are needed,—the man must love his work and have ability to perform it.

There is no royal road either to success or learning; the nearest approach to such a thoroughfare may be found in indefatigable study and reflection.

A smooth sea never made a skilful mariner. Neither do uninterrupted prosperity and happiness qualify a man for usefulness. The storms of adversity, like the storms of the ocean, arouse the faculties, and excite invention, prudence, skill, and fortitude.

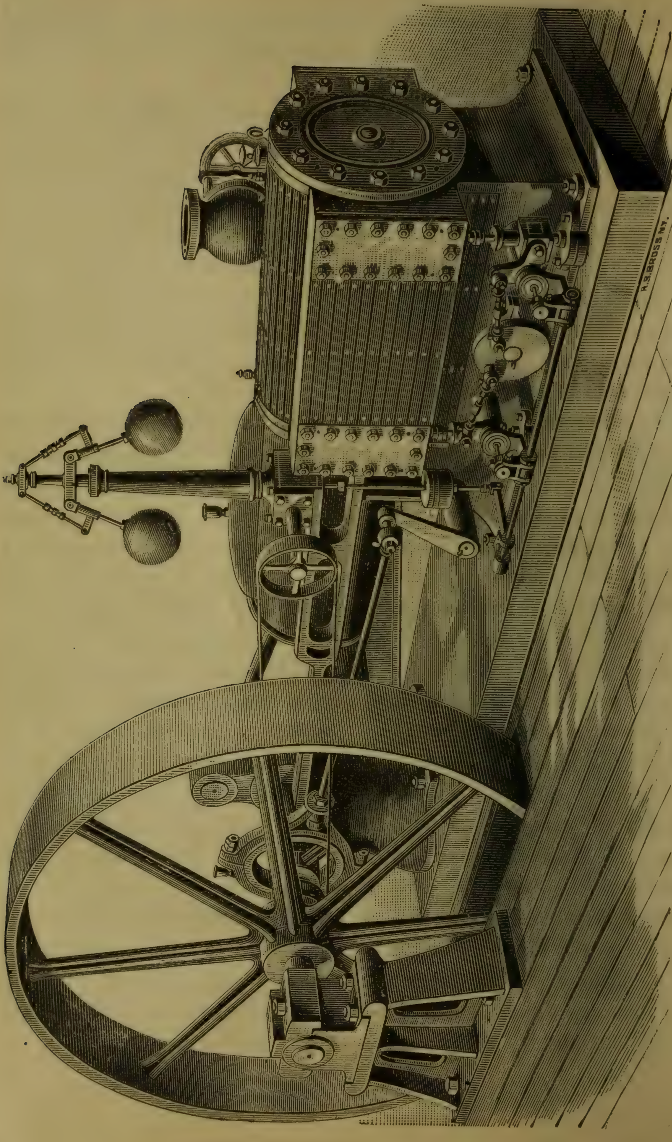
The very nature of steam engineering calls for superior intelligence in those on whom depend the care and management of steam machinery. Engineers should, therefore, prepare themselves for any casualty that may arise, by considering possible cases of derangement, and deciding in what way they would act should certain accidents occur.

The strength, perfection, and durability of steam machinery at the present day would seem to insure perfect safety, and yet accidents occur when least expected, for which no amount of mechanical skill or forethought could provide. It is in such cases that the coolness, determination, and decision of the engineer may avert a great calamity.

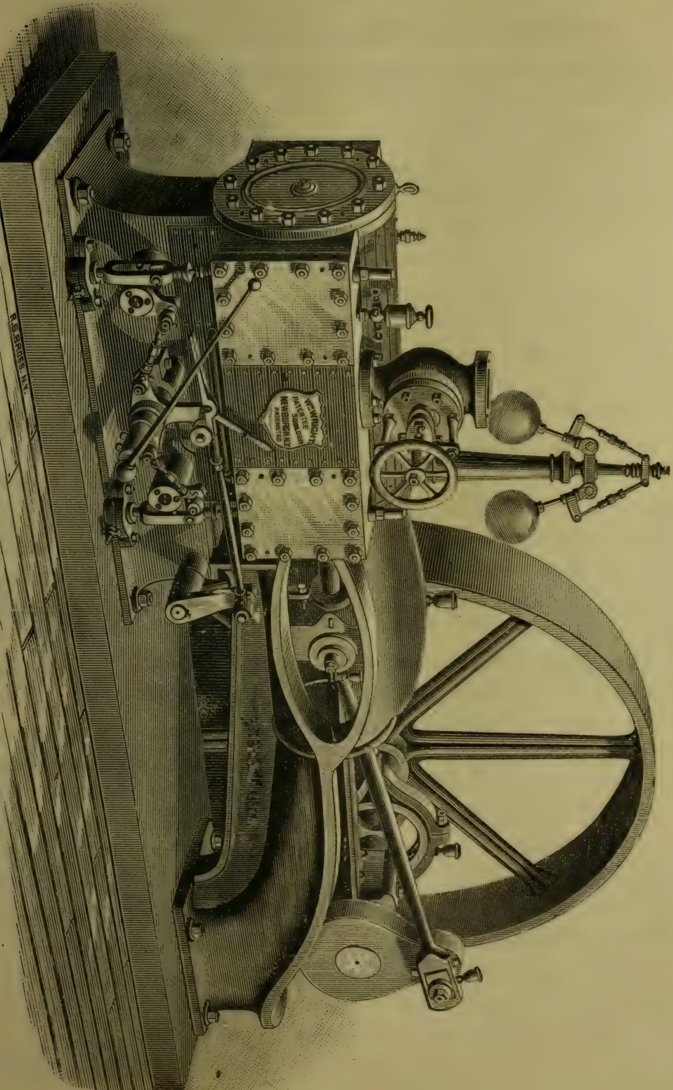
The wonderful increase in the size and speed of steamships and locomotives renders it absolutely necessary that, with a proper regard for life and property, they should be in charge of men of well-ascertained mental and physical abilities, as inevitably, sooner or later, at a critical moment, incapable men will be found wanting, and the most serious consequences result from their incapacity.

Risks of collision, of stranding, of fire, in fact, all risks pertaining to steamships, might be very much diminished if they were placed in charge of intelligent men.

The course to be pursued in an emergency must have reference to particular engines, as no general rules can be given, and every engineer should decide on certain measures to be adopted in any emergency in which he may be called upon to act, where everything may depend upon his energy and decision.



Exhaust Side of Wright's Automatic Cut-Off Engine.



Steam Side of Wright's Automatic Cut-Off Engine.

WRIGHT'S AUTOMATIC CUT-OFF ENGINE.

The cuts on pages 36 and 37 represent a front and back view of Wright's Automatic Cut-off high-pressure engine, the bed-plate or housing of which, as will be observed, is radically different in design and general appearance from any other in use in the country. The ordinary guides are dispensed with, and a guiding-cylinder, which is bored out on a line with the centre of the steam-cylinder, for the direction of movement of the cross-head, substituted. There are lateral openings in the sides of the guiding-cylinder, through which easy access to the cross-head and piston-rod may be had. From the front of the guiding-cylinder to the point where it meets the base, the frame is made in the form of an inclined concavo-convex trough of sufficient depth to permit the free movement of the connecting-rod. The trough has the upper edge of one side continued in a plane coinciding with the centre of the cylinder, from the latter to the enlargement formed to receive the bearing of the crank-shaft. The opposite side of the trough extends from the guiding-cylinder, with a gradually descending curve, to the base, into the upper portion of which it gracefully merges.

The steam-cylinder rests on a separate bed or foot, which sustains all the bearings for the valve-gear, and which is placed on a level with the pillow-blocks and main bed-plate, to which it is securely bolted and dowelled. There are four valves, two steam and two exhaust, which are of the gridiron or multiport-slide pattern. They work vertically in chests cast upon the cylinder, the two upon the front being for the induction cut-off and expansion, and those on the back for the eduction or exhaust. The motion for the steam- and exhaust-valves is derived from a single eccentric, which is so arranged as to give a quick movement to the valves in opening, and a slow movement when lapping. The stems of the steam-valves are connected with yokes having at their lower ends dash-pots, which act as guides. These yokes are operated by steel slides protruding through the ends of hollow rocker-arms, and acting upon the swinging toes held in the yokes.

The slides have a diagonal slot, in which works a feather on a rod, which has a longitudinal movement through the hollow rocker-arms with which the governor is connected. By this longitudinal motion through the diagonal feather and slot the slide is automatically set, to engage the swinging toes, more or less, according to requirements, to give the valve its proper lift, and release it on the chord of an arc.

The governor is of a kind peculiarly adapted to these engines, as it is very powerful and sensitive. It rests on a bracket, or shelf, cast on the side of the bed-plate; the rod being connected with a lever which is fastened to the governor-shaft. This shaft carries two forked arms, which take hold of the small rods running through the hollow rocker-shafts. These rods are enlarged at their ends, where they carry the adjustable slides which operate the steam-valve yokes.

These engines are made of the best material, and fitted and finished in the most thorough and workmanlike manner. The pistons are fitted with self-adjusting steam packing-rings, the lower half of piston body, between the packing-rings, being fitted with a brass shoe, which carries the weight of the piston, and can be so accurately adjusted that the piston must move central with the bore of the cylinder. The piston-rods, wrist- and crank-pins, and valve-rod are made of steel, and the boxes of the best machine brass. The connecting-rods and crank-shafts are made of the best hammered iron, and the pillow-blocks are lined with anti-friction metal. All the rubbing, revolving, and vibrating surfaces are of ample proportions, and fitted with great accuracy and precision.

The Wright Engine has undergone more changes in design and general appearance, since it was first introduced to steam users, than any other engine in the country. The manufacturer seems to be under the impression that, however much he might alter, he always discovers new defects in his engine, and he appears to be governed more by complication and weight than by symmetry and convenience. Consequently, the Wright engines are heavy,

unsymmetrical, and expensive, and perhaps less economical than most other automatic cut-off engines.

Examination of Candidates.

The best aids to a candidate applying for an engineer's certificate are preparation, coolness, and self-possession. He must be prepared to answer all questions propounded to him promptly and without hesitation, thereby showing that he is master of the situation, as hesitation in answering questions will convey the idea that his knowledge of the subject to which they relate is limited. Hesitation will, in all probability, induce the examiners to make the examination more lengthy and rigid; and it frequently happens that examinations, that might be completed in an hour, occupy several hours, and in some instances several days. Consequently, a candidate must be prepared to demonstrate problems, on any subject embraced in the programme of examination, by formulæ of his own; even if formulæ and problems be given him, he must be prepared to demonstrate that his method is equally as correct, besides being more brief and simple. He must understand that no amount of assurance will supply the place of study, nor will an empty assumption of knowledge compensate for a defective preparation.

Necessary Qualifications of Candidates Applying for Appointments as Cadet Engineers in the U. S. Navy.

Application may be made by candidates, or their friends, to the Secretary of the Navy, stating age, date of birth, educational advantages, and satisfactory evidence of health and good moral character, which is placed on register; but neither the registering of names, nor priority of application, gives any assurance of appointment. The number of appointments is limited by law to twenty-five annually. Applicants must be not under sixteen or over twenty years of age, and not less than five feet high. Those whose applications have been favorably received will be notified

to appear for examination on the 5th of September, at the Naval Academy.

Candidates must be physically sound, well formed, and of robust constitution; and those who possess the greatest skill and experience in the practical knowledge of machinery (other qualifications being equal), shall have precedence for admission.

The board of examiners have the power of exercising discretion in the application of the above requirements to each individual case, rejecting no candidate who is likely to be efficient in the service and admitting no one who is likely to prove physically inefficient. Candidates once rejected by the board of examination are not allowed a re-examination.

Candidates will be rejected for any of the following causes: Feeble constitution; greatly retarded development; permanently impaired general health. All chronic diseases, viz.: Weak or disordered intellect; cutaneous and communicable disease; unnatural curvature of spine, torticollis, or other deformity; permanent inefficiency of either of the extremities, or articulations from any cause; epilepsy; impaired vision, or chronic disease of the organs of vision; hardness of hearing, or chronic disease of the ears; chronic nasal catarrh, ozæma, polypi, or enlargement of the tonsils; impediment of speech; indications of pulmonary disease; chronic cardiac affections; hernia; sarcocele; hydrocele; stricture; fistula, or hæmorrhoids; varicose veins of lower limbs, *scrotum*, or *cord;* chronic ulcers.

Every cadet, immediately after his admission, must supply himself with clothing, bedding, toilet articles, sanitary utensils, etc.; the cost of which is $120. He must also deposit $50 with the paymaster, to be expended in the purchase of books, etc.; for which he will be credited on the books. He will also, one month after his admission, be credited with the amount of his expenses in travelling from his home to the Academy; but if he resigns his appointment within one year after admission, he will be required to refund the amount advanced him for travelling expenses.

Examination in Grammar.

Give the possessive singular and the objective plural of mayor, journey, sky, she, strife, wife.

Answer.—*Possessive singular:* Mayor's, journey's, sky's, her or hers, strife's, wife's. *Objective plural:* Mayors, journeys, skies, them, strifes, wives.

Give the principal parts of smite, shed, lay, lie, drown.

Present.	*Imperfect.*	*Present Participle.*	*Past Participle.*
Smite,	smote,	smiting,	smitten.
Shed,	shed,	shedding,	shed.
Lay,	laid,	laying,	laid.
Lie,	lay,	lying,	lain.
Drown,	drowned,	drowning,	drowned.
Lie,	lied,	lying,	lied.

Compare many, cleanly, shy, little, elder, without using adverbs.

Answer.—Many, more, most; cleanly, cleanlier, cleanliest; shy, shyer, shyest; little, less, least; old, older, oldest, or old, elder, eldest.

Name the moods, and explain the use of each.

Answer.—*Infinitive* expresses being, action, or passion, in an unlimited manner, without person or number.

Indicative simply indicates or declares a thing.

Potential expresses power, liberty, or necessity of being, action, or passion.

Subjunctive represents being, action, or passion as doubtful or contingent.

Imperative is used in commanding, exhorting, entreating, or permitting.

"**If you blow your neighbor's fire,** don't complain if the sparks fly in your face." Parse the words in dark type.

Answer.—*If* is a conjunction, connecting the sentence, "don't complain if the sparks fly in your face" with "you blow your neighbor's fire."

Blow is an irregular, active, transitive verb, subjunctive mood, present tense, second person, singular, to agree with its subject, "you."

Neighbor's is a common noun, third person, singular number, common gender, possessive case, governed by "fire."

Don't complain—contraction for "Do not complain"—is a regular, active, intransitive verb, emphatic form, in the imperative mood, second person, singular, to agree with its subject, "you," understood, conjugated negatively.

Your.—Your is a personal pronoun, second person, singular, common gender, to agree with its antecedent, "you," possessive case, governed by "face."

Fire is a common noun, third person, singular number, neuter gender, and objective case, object of the verb "blow."

Spelling.

Commissary.	Identify.	Anchorage.	Adjacent.
Treasury.	Precedent.	Correspond.	Occupant.
Debtor.	Asylum.	Similar.	Weaken.
Counterfeit.	Levy.	Eccentric.	Commercial.
Alliance.	Perpetrate.	Susceptible.	Insensible.
Apparition.	Although.	Sufficient.	Concession.

Examination in Arithmetic.

Express 16 days, 12 hours, 47 minutes, 25 seconds as a fraction of 23 days, 3 hours, 30 minutes, 23 seconds, (lowest terms.)

	d.	*hrs.*	*min.*	*sec.*	
Answer.	23	3	30	23	= 1,999,823 seconds.
	d.	*hrs.*	*min.*	*sec.*	
	16	12	47	25	= 1,428,445 "

$$\frac{1428445}{1999823}$$

Give the rule for finding the cube root of any number, and illustrate by an example.

Answer.—*First.* Point off from right to left, if an integer or

whole number, and from left to right, if a decimal, in orders or places of three. *Second.* Ascertain the highest root of the first order, and place it to the right of the number, as in long division. *Third.* Cube the root thus found, and subtract it from the first order, and to the remainder annex the next order; then square the root already found, and multiply it by three, with two ciphers annexed, for a trial divisor; next find how often this divisor is contained in the dividend, and write the result in the root. *Last.* Add together the trial divisor, three times the product of the first figure of the root, by the second, with one cipher annexed, and the square of the second figure in the root; multiply this last sum by the last figure in the root, and subtract as above; to the remainder annex the next order, and proceed, as before directed, until all the orders are worked.

To find the $\sqrt[3]{493039}$.

$$
\begin{array}{rr|l}
 & & 493039(79 \\
 & 7 \times 7 \times 7 = & 343 \\
\hline
7 \times 7 \times 3 = & 14700 & 150039 \\
7 \times 9 \times 3 = & 1890 & \\
9 \times 9 = & 81 & \\
\cline{2-2}
 & 16671 & 150039 \\
\cline{3-3}
\end{array}
$$

To find the $\sqrt[3]{403583{\cdot}419}$.

$$
\begin{array}{rr|l}
 & & 403583{\cdot}419(73{\cdot}9 \\
 & 7 \times 7 \times 7 = & 343 \\
\hline
7 \times 7 \times 3 = & 14700 & 60583 \\
7 \times 3 \times 3 = & 630 & \\
3 \times 3 = & 9 & \\
\cline{2-2}
 & 15339 & 46017 \\
\cline{2-3}
73 \times 73 \times 3 = & 1598700 & 14566419 \\
73 \times 3 \times 3 = & 19710 & \\
9 \times 9 = & 81 & \\
\cline{2-2}
 & 1618491 & 14566419 \\
\cline{3-3}
\end{array}
$$

The cube of any number is that number multiplied by itself three times.

Give the rule for finding the square root of any number, and illustrate by an example.

Answer.—*First.* Point off from right to left, if an integer or whole number, and from left to right, if a decimal, in orders or places of twos. *Second.* Ascertain the highest root in the first order, and place it at the right of the number, as in long division. *Third.* Square this root, and subtract it from the first order; to the remainder annex the next order, and double the root already found, and place it to the left of this dividend. *Fourth.* Ascertain how often this divisor is contained in all but the final figure of the dividend, and place the quotient to the right of the root already obtained, and to the right of the trial divisor. *Fifth.* Multiply this divisor by the final figure in the root, and subtract as before; if the remainder after a division is negative, take a figure for the last figure in the root one less than before, and proceed as directed in *Fourth* and *Fifth.* In like manner proceed until all the orders have been worked.

To find the $\sqrt{590 \cdot 49}$.

```
        5,90·49(24·3
        4
   44)  190
        176
  483)  1449
        1449
```

To find the $\sqrt{\cdot 075625}$.

```
        07,56,25(·275
         4
   47)  356
        329
  545)  2725
        2725
```

Any number multiplied by itself is squared.

Define the terms logarithms and hyperbolic logarithms, and explain their use.

Answer.—The logarithm of a number is the exponent of the power to which it is necessary to raise a fixed number in order to produce the first number. The use of logarithms is to abridge numerical computations. The operations of multiplication, division, involution, and evolution are very much abridged by their use. Any power of a given number may be found by logarithms as follows: The logarithm of any power of a given number is equal to the logarithm of the number multiplied by the exponent of the power.

Example.—To find the fifth power of 9, logarithm 9 = 0·954243 × 5 = 4·771215, and the number corresponding to this is 59049. *Conversely.* Any root of any number may be found by logarithms as follows: The logarithm of the root of a given number is equal to the logarithm of the number divided by the index of the root.

Example.—To find the cube root of 4096, logarithm 4096 = 3·612360 ÷ 3 = 1·204120, and the number corresponding to this logarithm is 16.

Hyperbolic logarithms is a system of logarithms, so called, because the numbers express the areas between the asymptote and curve of the hyperbola. The hyperbolic logarithm of any number is the common logarithm of the same number in the ratio of 2·30258509 to 1, or as 1 to ·43429448.

Explain the terms geometry and trigonometry.

Answer.—Geometry is the science of position and extension; that branch of mathematics which has for its object the investigation of the relations, properties, and measurement of solids, surfaces, lines, and angles. Trigonometry is that branch of mathematics whose object it is to determine unknown angles, or sides of triangles, by means of others which are known; the art or science of measuring triangles. It also treats of the general relations existing between the trigonometrical functions of angles or arcs.

Give the meanings of the terms quotient, product, and problem.

Answer.—A *quotient* is the result of an operation in division; a *product* is the result of an operation in multiplication; a *problem* is a question requiring some unknown truth to be demonstrated.

Give the meanings of the terms axiom, theorem, proposition, corollary, and solution.

Answer.—An *axiom* is a self-evident truth; a *theorem* is a statement of a truth or principle which is to be demonstrated; a *proposition* is a term applied to a theorem or a problem; a *corollary* is an obvious consequence deduced from one or more propositions; a *solution* is the result arising from any mathematical proposition or calculation.

Give the names of the various triangles, their peculiarities, etc.

Answer.—A *triangle* is a figure having three sides and three angles; an *isosceles triangle* has two sides and the angles at the base equal; a *scalene triangle* has no two sides or angles equal; an *obtuse-angled triangle* has one obtuse angle in it; a *right-angled triangle* has one right angle in it; an *equilateral triangle* has all three sides and angles equal; an *acute-angled triangle* has one acute angle in it.

Examination in Geography.

Name the States which have any coast-line on the great lakes, between the United States and Canada, telling in each case what lake the State touches.

Answer.

New York,	Lakes Ontario and Erie.
Pennsylvania,	Lake Erie.
Ohio,	Lake Erie.
Michigan,	Lakes Erie, St. Clair, Huron, Michigan, and Superior.
Indiana,	Lake Michigan.
Illinois,	Lake Michigan.
Wisconsin,	Lakes Michigan and Superior.
Minnesota,	Lake Superior.

Where and on what waters are Buenos Ayres, Bordeaux, Belgrade, Jackson, Bombay? Tell which of the above are capitals of States, and of what States they are capitals.

Answer.—Buenos Ayres is on the Rio de la Plata, and is the capital of the Argentine Confederation, South America; Bordeaux is on the Garonne, in France; Belgrade is on the Danube, in "Turkey in Europe;" Jackson is on the Pearl River, and is the capital of Mississippi; Bombay is in Hindostan, Asia, on the coast of the Arabian Sea.

Define the source, direction, and mouth of the Ganges River, Clyde River, Pruth River, Santee River.

Answer. — The source of the river Ganges is in the Himalaya Mountains; its direction is south-east; its mouth is in the north-eastern part of Hindostan, and empties into the Bay of Bengal. The source of the Clyde is in the Lammermoor Hills, Scotland; flows in a north-westerly direction, and discharges into the Firth of Clyde. The source of the Pruth is in the eastern base of the Carpathian Mountains, in Austria; flows first east, then south-east, and finally south, emptying into the Danube. The Santee is formed by the junction of the Congaree and Wateree Rivers in the central part of South Carolina; flows south-east, and empties into the Atlantic Ocean.

Where is Mount Snowden, the Atlas Mountains, the Elburz Mountains, Mount Ætna, Mount Chimborazo?

Answer.—Mount Snowden is a peak in the Cambrian range of Mountains, in Wales. The Atlas Mountains are in the "Barbary States," in Northern Africa. The Elburz Mountains are in Northern Persia, and form part of the great Himalayan range. Mount Chimborazo is a volcano in the Andes range, in Ecuador, South America.

Name five islands of the Mediterranean, define their position, and state to what Power each belongs.

Answer. — The Balearic group, off the east coast of Spain, belongs to Spain. Corsica, off the west coast of Italy, belongs to

France. Sardinia, off the west coast of Italy, belongs to Italy. Sicily, off the south-west coast of Italy, belongs to Italy. Candia, or Crete, off the south coast of Greece, belongs to Turkey.

Examination in Natural Philosophy.

Define centre of gravity of a body. How can the position of the centre of gravity of an irregular body be determined?

Answer.—The centre of gravity of a body is the point through which the resultant of the weights of the several component particles of a body always passes. The centre of gravity of an irregular body may be determined, experimentally, by suspending it successively from any two points, and, after it has come to a state of rest, or is in equilibrium, drawing, by means of a plumb-line, the verticals through the points of suspension. The intersection of these lines will be the centre of gravity of the body.

If the specific gravity of iron is 7·8 and that of gold 19·4, find the weight, in water, of a substance composed of one pound of iron and one pound of gold.

Answer.— In this problem, first find the solid contents, in inches, of one pound respectively of iron and of gold, and add them. Then, if one cubic foot of water weighs 62·5 lbs., one cubic inch will weigh ·036. Then multiply the combined solid contents in inches, of one pound of iron and gold, by the solid content in inches of one cubic inch of water. Thus:

1 lb. of iron = 3·54 cubic in. of solid contents.
1 " " gold = 1·42 " " " "
4·96 cubic in. of solid contents of both.

Then $4·96 \times ·036 = ·17856$.
This product deducted from 2 lbs. will be

$$\begin{array}{r} 2·00000 \\ 0·17856 \\ \hline 1·82144 \end{array}$$

Then multiply the weight of 1 cu. ft. of water respectively by that of iron and gold, and dividing by 1728 we get

$$62{\cdot}5 \times 7{\cdot}8 = 457{\cdot}5 \div 1728 = 3{\cdot}54 \text{ cu. in.}$$
$$62{\cdot}5 \times 19{\cdot}4 = 1212{\cdot}5 \div 1728 = 1{\cdot}42 \text{ " "}$$
$$4{\cdot}96 \text{ " "}$$

A cask weighing 236 lbs. 4 oz. floats in a square cistern of water whose side is 2 ft. 6 in. On the removal of the cask, find how much the water will sink in the cistern, supposing a cubic foot of water to weigh 63 lbs.?

Answer.—The weight of cask is 236 lbs. 4 oz., or, (if expressed decimally,) 236·25 lbs. Side of cistern is 2 ft. 6. in., or 30 inches; this squared equals 900 inches. Then the weight of 1 cubic foot of water is to 1 cubic foot of water as weight of cask is to cubic feet of water displaced by the cask.

lbs. oz. ... *ft. in.* *ft. in.*
236 4 = 236·25 — cistern 2 6 × 2 6 = 900 cu. in.
lbs. *cu. ft.* *lbs.* ... *cu. in.*
As 63 : 1 : : 236·25 : 3·75 cu. ft. 3·75 × 1728 = 6450
and 6450 ÷ 900 = 7·2 in.

How high will a common pump raise oil having a specific gravity of 0·88, if it raise water 33 feet? How would this height be affected if the force of gravity was doubled?

Answer.—The specific gravity of water is expressed by unity. Then the specific gravity of oil is to the specific gravity of water as the height to which water is raised is to the height to which oil will be raised by the same pump.

·88 : 1 : : 33 : 37·5 ft.

How far would a solid of any material sink into a fluid denser than itself?

Answer.—On the principle that fluids become denser the deeper they go, a solid would sink in any fluid till it displaced a volume of it equal to its own density.

Suppose the specific gravity of mercury be 14 and that of iron 7. How far would a cubic foot of iron sink into the mercury?

Answer. — One-half of its volume, or, if in a normal position, one-half its depth.

A cube of cork, whose edge is 1 foot, floating vertically in water, sinks to the depth of 2·88 inches. Find its specific gravity.

Answer. — First find the cubic contents of the water displaced, by multiplying the depth to which the cork sunk by the length and breadth of the same. Thus, 2·88 × 12 = 414·72 cu. in.; then, if 1728 cu. in. = 62·5 lbs., 414·72 cu. in. = 15 lbs. Hence, as 62·5 : 1000 :: 15 : 240, specific gravity of cork.

Give the readings of Fahrenheit's thermometer which correspond to 110°, 10°, 29° Centigrade; also the readings of the Centigrade thermometer which correspond to 77° and 23° Fahrenheit. At what temperature Centigrade will these two thermometers have the same readings?

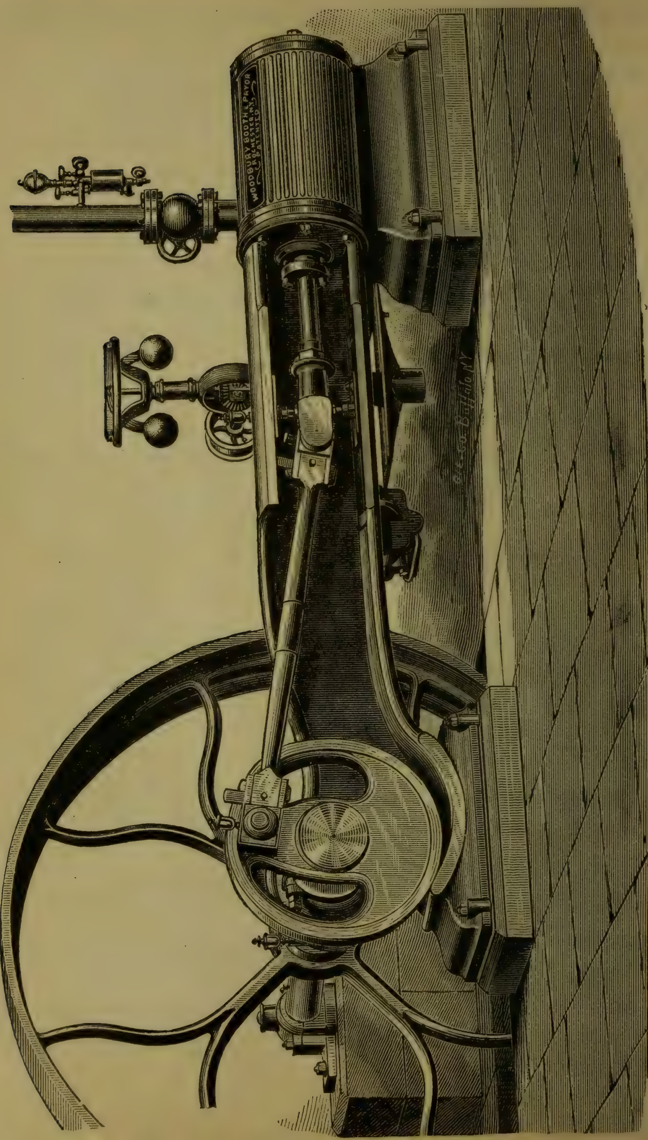
Answer.—To change Centigrade degrees to Fahrenheit degrees, multiply the Centigrade degrees by 9 and divide by 5, and to the quotient add 32. The result is Fahrenheit degrees.

C.	*C.*	*C.*
110°	10°	29°
9	9	9
5) 990	5) 90	5) 261
198	18	52·2
32°	32°	32°
230° F.	50° F.	84·2° F.

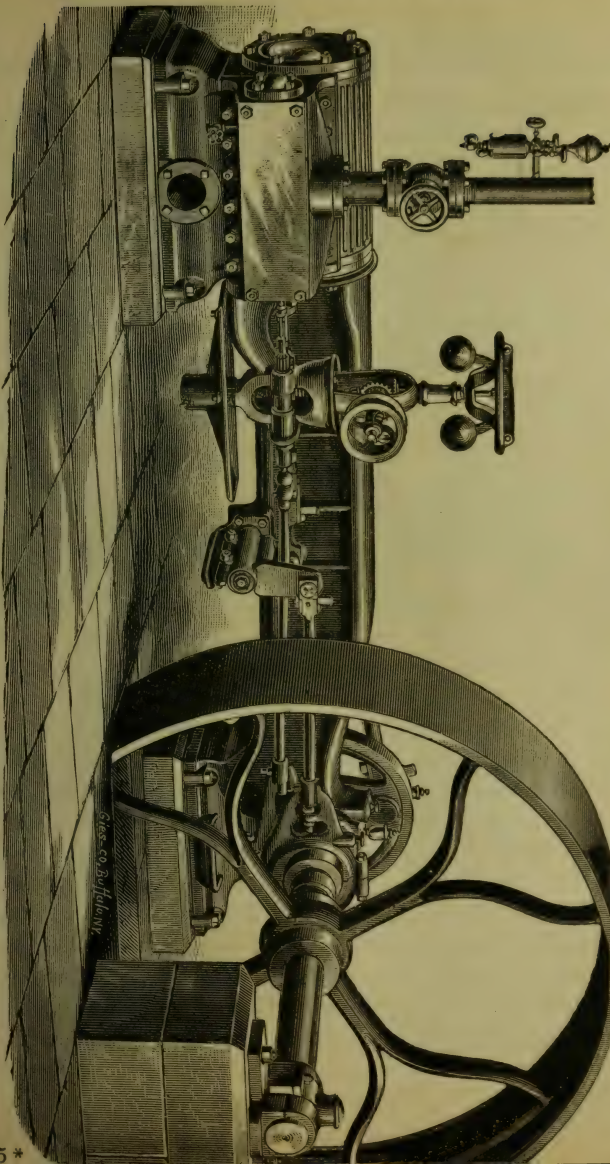
To change Fahrenheit degrees to Centigrade degrees, deduct 32° from Fahrenheit degrees; then multiply by 5 and divide by 9. The result is Centigrade degrees.

Explain the meaning of the term Conic Section.

Answer.—A conic section is a curve cut out of the surface of a right cone having a circular base, by a plane. The sections resulting from the cutting of a cone, are the triangle, the circle, the ellipse, the parabola, and the hyperbola; though the term conic section is confined to the last three.



Front View of Woodbury, Booth & Pryor's Automatic Cut-Off Engine.



Back View of Woodbury, Booth & Pryor's Automatic Cut-Off Engine.

Woodbury, Booth & Pryor's Automatic Cut-Off Engine.

The cuts on pages 52 and 53 represent Woodbury, Booth & Pryor's Automatic Cut-off Engine. It will be observed that it is built upon what is known as the girder or truss frame, a design which has been received with more favor, and more universally adopted by intelligent engineers, both in this country and Europe, than any other. In fact, it is fast superseding all previous forms of bed-plates. This perhaps arises from the fact that it is the best disposition that can be made of a certain quantity of material, to insure strength and rigidity without extra weight, as the centre line of the frame coincides with the plane of the line of the strains and with the centre line of the engine. It is faced at one end to receive the false head between the frame and cylinder, the piston-rod stuffing-box being cast with the head. The cylinder rests on a handsomely designed pedestal, the other end of the frame containing the back leg and main pillow-block bearing.

Valve-Gear.—The main valve is an ordinary D slide-valve, an arrangement which, since the advent of the steam-engine, in consequence of its simplicity of design, positiveness of action, moderate first cost, non-liability to become deranged and get out of order; that it performs the double function of admission and release, as well as from the ease and convenience with which it can be repaired or removed, has held its own against all new innovations in steam-valves and valve-gear. Fig. 1 shows a horizontal section through its centre, and, as will be observed, it is driven by an eccentric on the main shaft through the intervention of a rocker-arm. The cut-off valve, a section of which may be seen in Fig. 2, is cylindrical in form, and works in a small cylinder attached to the back of the main valve, which is cast in one piece with it. In consequence of the arrangement of the ports, it is perfectly balanced; and, owing to its having diagonal admission edges, with ports to correspond, by rolling it slightly in its seat it will cut off at any point between zero and three-quarter stroke, according to requirements.

The cut-off eccentric rod connects with the slide working in the bracket by means of a ball-and-socket joint, which allows the valve to rotate in its seat, more or less, according to the requirements of the load and pressure. The rotation, which never exceeds one-quarter of a revolution of the valve, is accomplished by a seg-

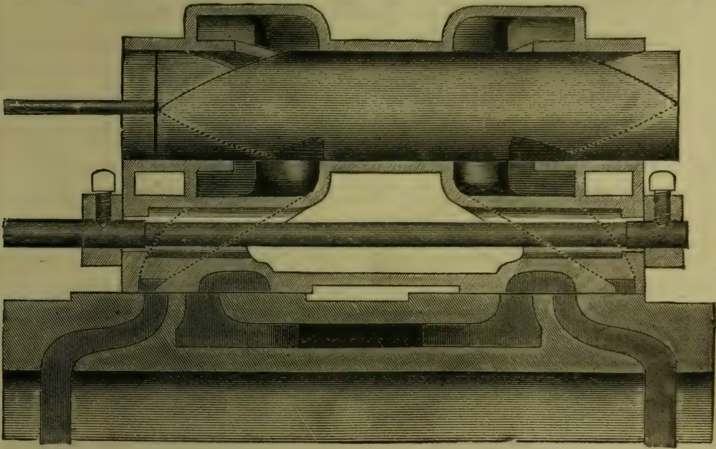


Fig. 1.

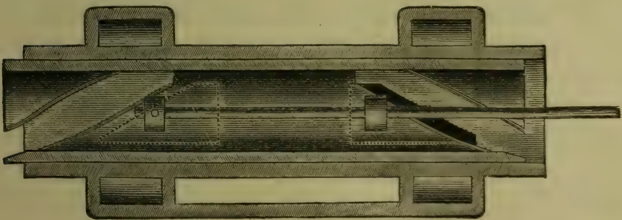


Fig. 2.

ment on the cut-off valve-slide, working into a rack attached to the governor spindle. The slide is a round steel rod, having a number of grooves milled in it lengthwise, sliding through corresponding tongues in the segment. This arrangement places the cut-off at all times under complete control of the governor, and, as the rolling movement of the cut-off is combined with the sliding

of the valve in its seat, it offers but very slight resistance, and induces less friction, even, than that of an ordinary throttling-valve. These cut-offs embody the following advantages: Simplicity of construction, positiveness of action, an entire absence of releasing gear, non-liability of derangement, freedom from violent shocks, accuracy in cutting off, and uniformity of speed.

The Governor.—The governor is of peculiar design; the arms extend across the centre, and have their point of suspension on the opposite side of the ball, which insures great sensitiveness, as they have a very extensive range of movement with a slight variation of speed. Besides, the governor has comparatively no work to perform, as the rolling of the balance cut-off while sliding lengthwise in its seat requires but a very small amount of force. In fact, the governor has nothing to do but to regulate the speed, without being hampered by the movement of cut-off valves, that have large rubbing surfaces exposed to boiler pressure.

The arrangement for the admission and release of the steam to and from the cylinder automatically, embodies the right principle by which to work steam expansively and economically. There are numerous industries in which uniform speed is not only a desideratum, but a necessity. It must not be understood, however, that all automatic cut-offs and governors are capable of producing satisfactory and economical results, as many of them in use at the present day fail to meet these requirements. The governors of these engines are provided with a dash-pot, which acts as a cushion, and obviates the jerking so common in many governors, and which is so detrimental to the uniform working of an engine.

Some of the advantages of the Woodbury, Booth & Pryor engines are, that they are handsome in design, and convenient and simple in arrangement; that they are built of the best materials, fitted with the greatest care, and steel and composition metals are freely used on parts subjected to the most wear, thus insuring durability as well as reducing the cost of maintenance to a minimum; that there is secured the highest degree of economy and the closest approximation to uniform speed; that the ports are dropped

sufficiently low to drain the cylinder, the water of condensation being drawn from the steam-chest instead of being forced through the cylinder; that owing to the positive valve-motion they can be run at a high rotative speed; that provision is made for catching all the drip oil and water; that the crank-pins have a hollow cellar or receptacle in the centre for the purpose of containing an auxiliary lubricant, which will melt and prevent the pin from cutting should the oil-cup be neglected or fail to feed; and that all wearing parts can be easily and accurately adjusted.

In fact, these engines seem to embody all the good points in the best descriptions of engines, and many others which are peculiar to themselves, which render them equal, if not superior, to any other automatic cut-off engines in the market. They are manufactured by Woodbury, Booth & Pryor, Rochester, N. Y.

Qualifications of Candidates for the U. S. Revenue Service.

First.—Candidates for appointments as second assistant engineers must not be less than twenty-one nor more than thirty years of age; they must be of good moral character and correct habits; they must have worked not less than eighteen months in a steam-engine factory, or served the same period as an engineer on board of a steamer having a condensing engine; they must also produce favorable testimonials from the superintendent of the machine-shop, or chief-engineer of the steam-ship, as to their ability.

Second.—They must be able to describe and sketch the different parts of marine steam-engines and boilers, and explain their uses and mechanical movements, the method of putting them in operation, regulating their action, and guarding against danger.

Third.—They must be fair arithmeticians and have a knowledge of rudimentary mechanics, be capable of writing a fair, legible hand, and have some knowledge of chemistry, particularly of combustion and corrosion.

Fourth.—Candidates who excel in practical experience and

professional skill will be given the preference, both in admission and promotion.

Fifth.—Any candidate producing a false certificate of age, time of service, or character, or making a false statement to the board of examiners, will be dropped from the list.

Standard of Examination for Assistant Engineer in the U. S. Revenue-Cutter Service.

First Assistant Engineer.

First.—They must pass before the board of examiners a thorough examination upon the subjects prescribed for second assistant engineers, and be able to explain the principles, peculiarities, functions, and uses of the different kinds of valves and valve-gear, as applied to marine steam machinery.

Second.—They must understand the construction, principles, peculiarities, and uses of the various mechanical arrangements employed in working steam expansively.

Third.—They must understand the construction of the marine boilers in most general use, their attachments, and the functions and uses of the same.

Fourth.—They must be able to explain the most general causes of derangement in the operation of air- and feed-pumps and pipes, and the most practicable method of preventing and remedying them.

Fifth.—They must have a knowledge of the chemical and mechanical causes which induce the formation of scale in steam-boilers, and the most practicable method of preventing and removing the same.

Sixth.—They must be acquainted with the general construction, principles, peculiarities, and uses of the different kinds of surface-condensers in present use.

Seventh.—They must be able to calculate the loss induced by blowing off, for the purpose of keeping the water in the boilers at a uniform degree of saturation, and understand the principles

of the various instruments employed to determine the water's saturation, as well as the method of graduating them.

Eighth.—They must understand the principles, most practicable limits, and advantages of working steam expansively, and be able to calculate the same.

Ninth. — They must have a knowledge of the construction of the indicator, know how to apply it, and intelligently explain its diagrams.

Tenth. — They must be acquainted with the construction and the principles on which the action of steam- and vacuum-gauges is based, and the causes of their derangement.

Eleventh. — They must have experience in building, erecting, and repairing steam machinery.

Examinations for the Mercantile Marine Service.

The following are among the questions most generally asked by examiners of engineers who apply for license in the mercantile marine service.

How long have you served as a fireman?

How long have you served in the engine-room at sea, and in what capacity?

With what description of engines have you served at sea — paddle or screw, jet-condensing, surface-condensing, or non-condensing engines, compound, trunk, inverted cylinder, or horizontal engines?

What size were the engines?

Explain the difference between condensing and non-condensing engines in principle and in point of economy.

In case the pump fails to work, what course would you adopt? and what are the most general causes of the failure of lift, or suction-bilge, or steam-pumps failing to act?

If the pumps fail to work, the water be low, and you are in danger of being driven on a lee-shore, what course would you adopt?

What is the object of braces in a steam-boiler? Which are preferable, a few large ones, or numerous small-sized ones?

What parts of a marine engine are most likely permanently to disable the ship in case of breakage?

Demonstrate by example (taking your own data) the safe working and bursting pressure of a boiler.

Demonstrate by example (taking your own data) the load necessary to be placed on the lever of a safety-valve for a given pressure of steam.

With what description of boilers have you served at sea — wet-bottomed, dry-bottomed, multi-tubular, sectional or flue, water- or fire-tube boilers?

What engine defects have come under your notice at sea? What caused those defects? How were they remedied? Give the names of the steamers.

What boiler defects have come under your notice at sea? What caused those defects?

How were they remedied? Give the names of the steamers.

Qualifications of Stationary Engineers.

In locations where the law requires persons having charge of stationary steam-engines to procure certificates of competency, the examination generally embraces the following subjects:

First. — Whether the candidate has charge of an engine at present. If not, where he had charge of one last. If he has a recommendation from his last employer. What was the size of the engine of which he had charge, diameter of cylinder, stroke, travel of piston, pressure as shown by the steam-gauge, etc., and what power such an engine was capable of developing.

Second. — Whether the engine was condensing or non-condensing, horizontal, vertical, inclined, oscillating, trunk, or beam, and the difference between a condensing and a non-condensing engine.

Third. — Whether the engine was automatic, cut-off or slide, throttling- or poppet-valve, and the difference between an automatic cut-off and slide-valve throttling-engine.

Fourth. — What he would consider his first duty on entering

the engine-room after being absent, or on taking charge of an engine and boiler for the first time. How he would proceed to set a slide-valve. How he could tell, without examining the valve, whether the engine was exhausting regularly or not, and what he would do before starting an engine, if it had been standing still for some time, particularly in cold weather.

Fifth.—Whether the boilers he had charge of were plain cylinder, flue, tubular, tubulous, or fire-box; whether they were internally or externally fired; and whether, on any occasion, he ever fired up under a boiler and afterwards discovered there was insufficient water in it.

Sixth.—What advantages and disadvantages do plain cylinder, flue, tubular, tubulous, and fire-box boilers possess over each other in point of economy of fuel, efficiency, safety, durability, and space?

Seventh.—How often should a boiler be cleaned, and how should it be managed before cleaning? How often ought boilers to be examined, and what should be the character of such an examination? What are the object and nature of the different tests now employed, and their effect on the boiler?

Eighth.—What course he would pursue in case the feed-water was cut off for a short time, or what would he do if cut off for an indefinite period, or from any cause became dangerously low; how he would proceed if obliged to stop his engine when steam was blowing off at the safety-valve and a heavy fire in the furnace; and how he would regulate his fire, when starting to raise steam, with cold water in the boiler.

Ninth.—The difference in the strains to which the shell, flues, tubes, crowns, and other parts of steam-boilers are subjected, as well as those which the longitudinal and curvilinear seams have to bear; also the difference in strength between single- and double-riveted seams, the loss induced by punching or drilling the holes for the rivets, and the difference in strength between punched and drilled holes, and between hand and machine riveting.

Tenth.—The diameter and length of the boiler of which he

had charge last; the diameter and length of the tubes or flues; the thickness of the shell; the number of square feet of heating surface in it — also of grate surface; the area of the safety-valve, and whether the boiler was single or double riveted.

Eleventh.—The safe working- and bursting-pressure of boilers, single or double riveted, of different diameters and of different thicknesses of iron; the proportions of grate and heating surfaces that would be capable of generating sufficient steam to develop a horse-power; and the orifice of safety-valve that would liberate that quantity of steam, provided that all other means of escape were closed.

Twelfth.—A demonstration, from his own data, of the weight necessary to place on the safety-valve for a given pressure when the length of the lever and the area of the safety-valve are known; also the pressure required to lift a certain weight, with a given length of lever and area of valve.

Thirteenth. — What are the most probable causes of lift or suction, force or boiler feed-pumps failing to work?

Locomotive Engineers.

Locomotive engineers are not required to furnish evidence that they possess any theoretical knowledge; nor are they required to pass any examination. They are all employed in the first place as firemen or brakemen, and their promotion to the charge of a locomotive depends on their sobriety, industry, and endurance. On most railroads they are required to fire from two to three years; after which, if they give evidence of sufficient capacity and carefulness, they are generally placed in the repair-shop or roundhouse for one year, to enable them to learn the use of tools, but more particularly to make them acquainted and familiar with the construction of the locomotive engine, and the manner of taking its machinery apart and putting it together again.

If, at the end of three or four years, he has conducted himself properly, and given sufficient evidence of his knowledge of the construction of a locomotive engine and its management to make

a good engineer, he is promoted to a *third-class* engineer. After one year's trial as third-class engineer, if he still gives evidence of capacity and carefulness, he is advanced to the position of second-class. If, after the expiration of one year as a second-class engineer, he is qualified in every way for a first-class engineer, he is advanced to that grade; but if not found competent, he is considered out of the regular order of promotion.

Steam.

Steam is an elastic fluid resulting from the combination of heat with water, and, when the steam is not in contact with the water from which it is formed, it follows the same general law as all other gases. This law is as follows: All gases expand by heat $\frac{1}{459}$th part of their volume for every degree Fah., while their elastic pressure remains unaltered, and so long as the temperature of a gas remains unaltered, its elastic pressure will vary inversely to the volume. Steam is of several kinds. *Surcharged steam* is steam heated to a temperature higher than is due to its pressure. *Saturated steam* is steam which, in contact with the fluid from which it is formed, has brought with it a proportion of moisture.

Supersaturated steam is steam in which there is more water mingled in the form of minute spray than is generally contained in saturated steam, which is called the water of supersaturation.

The temperature of the steam is always equal to that of the water from which it is formed, and the elastic force of steam formed is equal to the pressure under which it is formed. The elastic force of steam, barometer at 30°, at 212° Fah., is one atmosphere, or 14·7 lbs. per sq. inch; while at 250° Fah. its elastic force is two atmospheres, or 29·4 lbs. per sq. inch. This includes the pressure of the atmosphere.

If the mercury be in a vacuum, the pressure of steam due to a temperature of 212° Fah. will equal 30 inches, and for a pressure due to a temperature of 250° Fah. it will equal 60 inches; but if the mercury be exposed to the atmosphere, the pressure due to 250° Fah. will only equal 30 inches of mercury, and for 212°

Fah. there is no indication by a mercury gauge, as steam at 212° just balances the atmosphere.

The volume of steam is the space which it occupies. At 15 lbs. pressure above atmosphere, its volume is 883, and at 30 lbs. its volume is 610 times the space it occupied in the shape of water.

Surcharged steam is not indicated by the steam-gauge, as the steam-gauge only shows the existence of pressure; but it may be indicated by a thermometer gauge, or by a fusible plug.

If the proper relation of the temperature between the steam and water be disturbed, a violent ebullition or foaming will generally take place, and will continue till the natural relation is restored. This foaming is a source of danger to the engine and boilers.

The total heat of steam at 212° Fah. is 1202°, of which 990° are latent heat, which is heat that is neither sensible to the touch, nor can it be indicated by the thermometer. The existence of this latent heat in water, while in the form of steam, may be proved by the following illustration: If 5½ lbs. of water, at 32° Fah., are placed in a vessel communicating with another, in which water is kept at 212° Fah., and kept there till the former reaches a temperature of 212° Fah., and then weighed, it will be found to weigh 6½ lbs., showing that 1 lb. of water has been added to the 5½ lbs. in the form of steam. This pound of water, received in the form of steam had, when in that form, a temperature of 212° Fah. It still possesses the same temperature of 212° Fah., showing that it has parted with 5½ times the number of degrees of temperature between 32° and 212°, which is 180, and $5\frac{1}{2} \times 180 = 990°$. This heat was combined with the steam, but not being sensible to the thermometer is called latent; in this connection 5½ is taken as a convenient number.

If we observe the time that a certain amount of heat takes to raise water from 32° to 212°, no matter what the time may be, it will take 5½ times as long for the same heat to evaporate the same amount of water. It follows, that to evaporate water under the pressure of the atmosphere requires 5½ times as much heat as

would be necessary to raise the same amount of water from 32° to 212°.

A pound of steam in passing from a liquid at 212° to steam at 212° receives as much heat as would be sufficient to raise it through 990°, if that heat, instead of being latent, had been sensible, and 990° + 212 = 1202° is the whole amount of the heat in steam.

The latent heat of steam is found by deducting its sensible heat from 1202°.

TABLE

SHOWING THE INCREASE OF SENSIBLE AND THE DECREASE OF LATENT HEAT IN STEAM, ACCORDING TO PRESSURE.

GROSS PRESSURE.	SENSIBLE HEAT.	LATENT HEAT.	RELATIVE VOLUME.
15 lbs.	212°	966·2°	1669
30 "	251°	939·0°	881
45 "	275°	922·7°	608
60 "	294°	909·2°	467
75 "	309°	898·5°	381
90 "	320°	891·3°	323

Heat in steam becomes latent whenever a change takes place in the temperature; then the heat produces the change, but does not raise the temperature.

The heat necessary to generate steam, instead of being 212°, must be 966 + 212° = 1178°; therefore, the coal consumed and the water necessary to condense the steam must be 5½ times as great as they would be, if the heat were all sensible instead of latent, which, it will be observed, very materially affects the economy of the steam-engine.

The amount of water necessary to condense a certain quantity of steam may be found as follows: If a cubic inch of water produces a cubic foot of steam, and the latent heat of steam at 212° be taken at 990°, or, in other words, if the cubic foot of steam be supposed to contain as much heat in the latent form as would raise the temperature of the cubic inch of water, if it could be

prevented from expanding, then 990°, the sum of the latent heat, will be represented by 1202°. The temperature of the water discharged by the air-pump is about 100°, which, deducted from 1202°, leaves 1102°, which must be taken up by such a quantity of cold water that its temperature will not rise above 100°. If the temperature of the injection-water be 50°, then the difference between that and 100° is 50°, which is available for the absorption of heat, and $\frac{1102°}{50°} = 22{\cdot}1$, which is the number of times the injection-water must exceed the quantity of water in the steam. Inasmuch as the injection-water is seldom so cold, a much larger proportion of injection-water is usually required.

Steam which has any elastic force not exceeding that of one atmosphere is termed "low pressure" steam, and "high pressure" is only low pressure steam compressed into a smaller space.

Surcharged steam will affect the vacuum of an engine on account of the undue amount of heat which it contains; therefore, the amount of injection-water must be increased to take up this extra heat, and keep the condensed water at the proper temperature.

The steam from salt water is fresh, because no salt is carried away in the steam when evaporation from salt water takes place; and when the water is all evaporated, the original salt will be found in the vessel.

The difference in volume between water and steam at atmospheric pressure is 1669; that is, a given quantity of water, when converted into steam, will occupy 1669 times that which the water did. One cubic foot of steam, at atmospheric pressure, weighs ·038 of a pound.

A steam-jacket is a hollow casing surrounding the cylinders of steam-engines, into which the exhaust steam is admitted in its escape from the cylinder. Its object is to preserve a uniform temperature, and to prevent radiation and condensation. The benefit to be derived from its use, in any case, is an unsettled question among engineers.

TABLE

SHOWING THE EFFLUENT VELOCITY WITH WHICH STEAM, AT DIFFERENT PRESSURES, WILL FLOW INTO THE ATMOSPHERE, OR INTO STEAM AT A LOWER PRESSURE.

PRESSURE ABOVE THE ATMOSPHERE.	VELOCITY OF ESCAPE PER SECOND.	PRESSURE ABOVE THE ATMOSPHERE.	VELOCITY OF ESCAPE PER SECOND.
Pounds.	Feet.	Pounds.	Feet.
1	540	50	1736
2	698	60	1777
3	814	70	1810
4	905	80	1835
5	981	90	1857
10	1232	100	1875
20	1476	110	1889
30	1601	120	1900
40	1681	130	1909

Rule for Finding the Amount of Gain derived from Working Steam Expansively.

Divide the length of the stroke in feet by the cut-off, $\frac{1}{3}$, $\frac{1}{4}$, $\frac{1}{2}$, as the case may be; then find on the table on page 68 the hyperbolic logarithm nearest to that of the quotient, to which add 1. This sum will give the ratio of gain.

Example. — Suppose 50 pounds per square inch to be the initial pressure; length of stroke, 10 feet; cut-off, $\frac{1}{4}$; find the mean pressure.

$10 \div 2{\cdot}5 = 4$. The hyperbolic logarithm of 4 is 1·38629, which, with 1 added, becomes 2·38629, which is the ratio of gain.

$4 : 2{\cdot}38629 :: 50 = \dfrac{2{\cdot}38629 \times 50}{4} = 29{\cdot}82862$ lbs. mean or average pressure.

If a given quantity of Steam, the expansive power of which, at full pressure, is represented by 1, be admitted to a cylinder of a certain size, and cut off when the piston travels through $\frac{1}{2}$ of the stroke, its effect will be raised by expansion to 1·69; if cut off at $\frac{1}{3}$, the effect will be 2·10; at $\frac{1}{4}$, 2·39; at $\frac{1}{5}$, 2·61; at $\frac{1}{6}$, 2·79; at $\frac{1}{7}$, 2·95; at $\frac{1}{8}$, 3·08; but the expansion cannot be carried beneficially as far as $\frac{1}{8}$ in all classes of engines.

TABLE

OF HYPERBOLIC LOGARITHMS TO BE USED IN CONNECTION WITH THE ABOVE RULE.

No.	Logarithm.	No.	Logarithm.	No.	Logarithm.
1·25	·22314	5·	1·60943	9·	2·19722
1·5	·40546	5·25	1·65822	9·5	2·25129
1·75	·55961	5·5	1·70474	10·	2·30258
2·	·69314	5·75	1·74919	11·	2·39789
2·25	·81093	6·	1·79175	12·	2·48490
2·5	·91629	6·25	1·83258	13·	2·56494
2·75	1·01160	6·5	1·87180	14·	2·63905
3·	1·09861	6·75	1·90954	15·	2·70805
3·25	1·17865	7·	1·94591	16·	2·77258
3·5	1·25276	7·25	1·98100	17·	2·83321
3·75	1·32175	7·5	2·01490	18·	2·89037
4·	1·38629	7·75	2·04769	19·	2·94443
4·25	1·44691	8·	2·07944	20·	2·99573
4·5	1·50507	8·5	2·14006	21·	3·04452
4·75	1·55814			22·	3·09104

Rule for finding the mean or average pressure in the cylinder of a steam-engine.

Divide the length of the stroke in inches (including the clearance) by the distance that the steam follows the piston before being cut off; the quotient will be the expansion the steam undergoes. Then find in the expansion column, in the following table, the number corresponding to it; take the multiplier opposite, and multiply the full pressure of the steam per square inch, as it enters the cylinder, by it. The product will be the average pressure.

Example.—Suppose the initial pressure be 70 lbs. per sq. inch and cut-off at half-stroke, the stroke being 3 ft.

Then 3 ft. $= 36$ in. $+ 0·5$ for clearance $= 36·5$.

Stroke $\frac{1}{2} = 18$ in. $+ 0·5$ " $= 18·5$.

Then $36·5 \div 18·5 = 1·97$, the relative expansion between 1·9 and 2. By referring to the table, the multiplier for 1·9 will be found to be 0·864, and the difference between that and the multiplier for 2 is 0·017. Hence, by multiplying 0·017 by ·07, and subtracting the product 0·011, the remainder, 0·86281, is the multiplier for 1·97. Therefore, $0·86281 \times 70 = 60·3967$ lbs. per sq inch, the mean effective pressure on the piston.

TABLE

OF MULTIPLIERS BY WHICH TO FIND THE MEAN PRESSURE OF STEAM AT VARIOUS POINTS OF CUT-OFF.

Expansion.	Multiplier.	Expansion.	Multiplier.	Expansion.	Multiplier.
1·0	1·000	3·4	·654	5·8	·479
1·1	·995	3·5	·644	5·9	·474
1·2	·985	3·6	·634	6·	·470
1·3	·971	3·7	·624	6·1	·466
1·4	·955	3·8	·615	6·2	·462
1·5	·937	3·9	·605	6·3	·458
1·6	·919	4·	·597	6·4	·454
1·7	·900	4·1	·588	6·5	·450
1·8	·882	4·2	·580	6·6	·446
1·9	·864	4·3	·572	6·7	·442
2·	·847	4·4	·564	6·8	·438
2·1	·830	4·5	·556	6·9	·434
2·2	·813	4·6	·549	7·	·430
2·3	·797	4·7	·542	7·1	·427
2·4	·781	4·8	·535	7·2	·423
2·5	·766	4·9	·528	7·3	·420
2·6	·752	5·	·522	7·4	·417
2·7	·738	5·1	·516	7·5	·414
2·8	·725	5·2	·510	7·6	·411
2·9	·712	5·3	·504	7·7	·408
3·	·700	5·4	·499	7·8	·405
3·1	·688	5·5	·494	7·9	·402
3·2	·676	5·6	·489	8·	·399
3·3	·665	5·7	·484		

TABLE

OF CONSTANT NUMBERS, BY WHICH TO ASCERTAIN THE AVERAGE PRESSURE OF THE STEAM AGAINST THE PISTON FOR DIFFERENT PRESSURES AND POINTS OF CUT-OFF, FROM $\frac{1}{4}$ TO $\frac{7}{8}$ OF THE STROKE.

Point of Cut-off.	Constant Number.	Point of Cut-off.	Constant Number.
$\frac{1}{4}$	·5965	$\frac{5}{8}$	·9188
$\frac{1}{3}$	·6995	$\frac{2}{3}$	·9370
$\frac{3}{8}$	·7428	$\frac{3}{4}$	·9657
$\frac{1}{2}$	·8465	$\frac{7}{8}$	·9919

Multiply the pressure in pounds, as shown by the gauge, by the constant number opposite the point of cut-off in the left column. The product is the average pressure.

TABLE

OF CONSTANT NUMBERS FOR FINDING THE REQUIRED "LAP" FOR SLIDE-VALVES, WHEN THE TRAVEL OF THE VALVE IS KNOWN.

Cut-off.....	$\frac{1}{2}$	$\frac{7}{12}$	$\frac{2}{3}$	$\frac{3}{4}$	$\frac{5}{6}$	$\frac{7}{8}$	$\frac{11}{12}$
Multiplier.	·354	·323	·289	·250	·204	·177	·144

Multiply the valve-stroke by the decimals opposite each point of cut-off.

There are two methods of applying the power of steam to the cylinders of steam-engines, one being to allow it to flow from the boiler to the cylinder through the whole length of the stroke, the other to cut off the supply when the piston has travelled a certain distance. The advantage of the latter over the former consists in the saving of fuel; which may be explained as follows: If steam be applied the full length of the stroke, the average pressure will be as the pressure per square inch on the piston; but if the steam be cut off at half stroke,—suppose the pressure to be 65 lbs. per square inch, when the pressure of the atmosphere is added,—there will be a mean equivalent, or average pressure, throughout the stroke of about 55 lbs. per square inch, being only 10 lbs. less than full pressure, or 16 per cent. of a loss in power, though only half the former quantity of steam has been used.

Steam-ports.—A term applied to the passages through which the steam enters the cylinder; they are generally $\frac{1}{16}$ the area of the piston, but vary for different boiler pressures and piston speeds, those of locomotives being about $\frac{1}{10}$, which have the largest ports of any class of engines. The area of the exhaust-port should be from $\frac{1}{8}$ to $\frac{1}{6}$ that of the cylinder. As a rule, the exhaust, when passing out of the cylinder after the first rush is over, should not have to travel faster than 100 feet per second; but with some designs of engines the velocity of the steam may be greater, without creating injurious back pressure. The form of the ports is immaterial, providing they are large enough to give admission to the amount of steam requisite to keep the pressure up to its initial point until it is cut off by the valve, and give free egress for its escape.

TABLE

SHOWING THE AVERAGE PRESSURE OF THE STEAM UPON THE PISTON THROUGHOUT THE STROKE, WHEN CUT OFF IN THE CYLINDER FROM $\frac{1}{3}$ TO $\frac{7}{8}$, COMMENCING WITH 10 POUNDS AND ADVANCING IN 5 POUNDS UP TO 55 POUNDS PRESSURE.

Steam cut off in the Cylinder.	Pressure in Pounds at the Commencement of the Stroke.									
	10	15	20	25	30	35	40	45	50	55
	Average Pressure in Pounds upon the Piston.									
$\frac{1}{3}$	7	10½	14½	17½	21	24½	28	31½	35	38½
$\frac{2}{3}$	9¼	14	18¾	23½	28¼	32¾	37½	42	46¾	51½
$\frac{1}{4}$	6	9	12	15	17¾	20¾	23¾	26¾	29¾	32¾
$\frac{1}{2}$	8½	12¾	17	21	25¼	29½	33¾	38	42¼	46½
$\frac{3}{4}$	9½	14½	19¼	24	28¾	33½	38½	43¼	48¼	53
$\frac{1}{5}$	5¼	7¾	10½	13	15½	18¼	20¾	23½	26	28½
$\frac{2}{5}$	7½	11½	15¼	19	23	26¾	30½	34½	38¼	42
$\frac{3}{5}$	9	13	18	22½	27	31½	36¼	40¾	45¼	49¾
$\frac{4}{5}$	9¾	14½	19½	23¾	29¼	34¼	39	44	49	53¾
$\frac{1}{6}$	4½	7	9¼	11½	14	16¼	18½	20¾	23¼	25½
$\frac{5}{6}$	9¾	14¾	19¾	24½	29½	34½	39½	44¼	49¼	54
$\frac{1}{7}$	4¼	6¼	8¼	10½	12½	14¾	16¾	18¾	21	23¼
$\frac{2}{7}$	6½	9½	12¾	16	19¼	22½	25¾	28¾	32	35¼
$\frac{3}{7}$	7¾	11¾	15¾	19¾	23¾	27¾	31½	35½	39½	43½
$\frac{4}{7}$	8¾	13¼	17¾	22¼	26¾	31¼	35½	40	44½	49
$\frac{5}{7}$	9½	14¼	19	23¾	28½	33½	38¼	42¾	47¾	52½
$\frac{6}{7}$	9¾	14¾	19¾	24¾	29½	34½	39½	44½	49½	54¼
$\frac{1}{8}$	3¾	5¾	7¾	9½	11½	13½	15¼	17¼	19¼	21¼
$\frac{3}{8}$	7¼	11	14¾	18½	22¼	26	29¾	33½	37	40¾
$\frac{5}{8}$	9¼	13¾	18¼	22¾	27½	32	36¾	41¼	45½	50½
$\frac{7}{8}$	9¾	14¾	19¾	24¾	29¾	34¾	39½	44½	49½	54½

TABLE

SHOWING THE AVERAGE PRESSURE OF THE STEAM UPON THE PISTON THROUGHOUT THE STROKE, WHEN CUT OFF IN THE CYLINDER FROM ⅓ TO ⅞, COMMENCING WITH 60 POUNDS AND ADVANCING IN 5 POUNDS UP TO 105 POUNDS PRESSURE.

Steam cut off in the Cylinder.	Pressure in Pounds at the Commencement of the Stroke.									
	60	65	70	75	80	85	90	95	100	105
	Average Pressure in Pounds upon the Piston.									
⅓	42	45½	49	52½	56	59½	63	66½	70	73
⅔	56¼	61	65½	70¼	75	79½	84½	89	93¾	98¼
¼	35¾	38¾	41¾	44¾	47¾	50¾	53¾	56¾	59¾	62¾
½	50¾	55	59¼	63½	67¾	72	76¼	80½	84¾	89
¾	57¾	62½	67½	72¼	77¼	82	87	91¾	96½	101¼
⅕	31¼	34	36½	39	41¾	44¼	47	49½	52¼	54¾
⅖	46	49¾	53½	57½	61¼	65	69	72¾	76½	80¼
⅗	54¼	58¾	63¼	67¾	72½	77	81½	86	90½	95¼
⅘	58½	63½	68½	73¾	78¼	83	88	92¾	97¾	102¾
⅙	27¾	30¼	32½	34¾	37¼	39½	41¾	44¼	46½	48¾
⅚	59	64	69	73¾	78¾	83¾	88¾	93½	98½	103½
1/7	25¼	27¼	29½	31½	33½	35¾	37¾	40	42	44
2/7	38½	41¾	45	48¼	51½	54½	57¾	61	64¼	67½
3/7	47½	51½	55¼	59¼	63¼	67¼	71¼	75¼	79	83
4/7	53½	57¾	62½	66¾	71¼	75¾	80	84½	89	93½
5/7	57¼	62	66¾	71½	76¼	81	85¾	90¾	95½	100¼
6/7	59¼	63¾	69¼	74¼	79	84	89	93¾	98[illegible]	103¾
⅛	23	25	27	28¾	30¾	32¾	34½	36½	38½	40½
⅜	44½	48¼	52	55¾	59½	63	66¾	70½	74¼	78
⅝	55¼	59¾	64¼	68¾	73½	78	82½	87¼	91¾	96½
⅞	59½	64½	69¼	74¼	79¼	84¼	89¼	94¼	99	104

TABLE

SHOWING THE AVERAGE PRESSURE OF THE STEAM UPON THE PISTON THROUGHOUT THE STROKE, WHEN CUT OFF IN THE CYLINDER FROM ⅓ TO ⅞, COMMENCING WITH 110 POUNDS AND ADVANCING IN 5 POUNDS UP TO 150 POUNDS PRESSURE.

Steam cut off in the Cylinder.	Pressure in Pounds at the Commencement of the Stroke.								
	110	115	120	125	130	135	140	145	150
	Average Pressure in Pounds upon the Piston.								
⅓	77½	80½	84	87½	91	94½	98	101½	105
⅔	103	107¾	112½	117	121¾	126½	131	135¾	140½
¼	65½	68½	71½	74½	77½	80½	83½	86½	89½
½	93¼	97¼	101½	105¾	110	114¼	118½	122¾	127
¾	106¼	111	115¾	120¾	125½	130¼	135¼	140	144¾
⅕	57¼	60	62½	65¼	67¾	70½	73	75½	78½
⅖	84¼	88	91¾	95¾	99½	103¼	107¼	111	115
⅗	99½	104¼	108¾	113¼	117¾	122¼	126¾	131½	135
⅘	107½	112½	117½	122¼	127¼	132	136¾	141¾	146¾
⅙	51¼	53½	55¾	58	60½	62¾	65	67½	69¾
⅚	108¼	113¼	118¼	123¼	128	133	137¾	142¾	147
⅐	46¼	48¼	50½	52½	54¾	56¾	58¾	61	63
2/7	70¾	74	77¼	80½	83½	86¾	90	93¼	96½
3/7	87	91	94¾	98¾	102¼	106¾	110¾	114¾	118¾
4/7	98	102½	106¾	111½	115¾	120¼	124¾	129¼	133½
5/7	105	109¾	114½	119¼	124	128¾	133½	138½	143¼
6/7	108¾	113¾	118¾	123½	128½	133½	138½	143½	148¼
⅛	42¾	44½	46¼	48	50	52	54¼	56¼	57¾
⅜	81¾	85½	89	92¾	96½	100¼	104	107¾	111½
⅝	101	105½	110¼	114¾	119½	124	128½	133¼	137¾
⅞	109	114	119	124	128¾	133¾	138¾	143¾	148¾

TABLE

SHOWING THE TEMPERATURE OF STEAM AT DIFFERENT PRESSURES, FROM 1 LB. PER SQUARE INCH TO 220 LBS., AND THE QUANTITY OF STEAM PRODUCED FROM A CUBIC INCH OF WATER, ACCORDING TO PRESSURE.

It is necessary to add the pressure of the atmosphere, 15 pounds, to the pressure on the steam-gauge, to correspond with the table.

Total Pressure of Steam in lbs. per Square Inch.	Correspond-ing Temper-ature of Steam to Pressure.	Cubic Inches of Steam from a Cubic Inch of Water according to Pressure.	Total Pressure of Steam in lbs. per Square Inch.	Correspond-ing Temper-ature of Steam to Pressure.	Cubic Inches of Steam from a Cubic Inch of Water according to Pressure.
1	102·9°	20868	29	249·6°	911
2	126·1	10874	30	251·6	883
3	141·0	7437	31	253·6	857
4	152·3	5685	32	255·5	833
5	161·4	4617	33	257·3	810
6	169·2	3897	34	259·1	788
7	175·9	3376	35	260·9	767
8	182·0	2983	36	262·6	748
9	187·4	2674	37	264·3	729
10	192·4	2426	38	265·9	712
11	197·0	2221	39	267·5	695
12	201·3	2050	40	269·1	679
13	205·3	1904	41	270·6	664
14	209·1	1778	42	272·1	649
15	212·8	1669	43	273·6	635
16	216·3	1573	44	275·0	622
17	219·6	1488	45	276·4	610
18	222·7	1411	46	277·8	598
19	225·6	1343	47	279·2	586
20	228·5	1281	48	280·5	575
21	231·2	1225	49	281·9	564
22	233·8	1174	50	283·2	554
23	236·3	1127	51	284·4	544
24	238·7	1084	52	285·7	534
25	241·0	1044	53	286·9	525
26	243·3	1007	54	288·1	516
27	245·5	973	55	289·3	508
28	247·6	941	56	290·5	500

TABLE

SHOWING THE TEMPERATURE OF STEAM AT DIFFERENT PRESSURES, FROM 1 LB. PER SQUARE INCH TO 220 LBS., AND THE QUANTITY OF STEAM PRODUCED FROM A CUBIC INCH OF WATER, ACCORDING TO PRESSURE.

It is necessary to add the pressure of the atmosphere, 15 pounds, to the pressure on the steam-gauge, to correspond with the table.

Total Pressure of Steam in lbs. per Square Inch.	Corresponding Temperature of Steam to Pressure.	Cubic Inches of Steam from a Cubic Inch of Water according to Pressure.	Total Pressure of Steam in lbs. per Square Inch.	Corresponding Temperature of Steam to Pressure.	Cubic Inches of Steam from a Cubic Inch of Water according to Pressure.
57	291·7°	492	85	320·1°	342
58	292·9	484	86	321·0	339
59	294·2	477	87	321·8	335
60	295·6	470	88	322·6	332
61	296·9	463	89	323·5	328
62	298·1	456	90	324·3	325
63	299·2	449	91	325·1	322
64	300·3	443	92	325·9	319
65	301·3	437	93	326·7	316
66	302·4	431	94	327·5	313
67	303·4	425	95	328·2	310
68	304·4	419	96	329·0	307
69	305·4	414	97	329·8	304
70	306·4	408	98	330·5	301
71	307·4	403	99	331·3	298
72	308·4	398	100	332·0	295
73	309·3	393	110	339·2	271
74	310·3	388	120	345·8	251
75	311·2	383	130	352·1	233
76	312·2	379	140	357·9	218
77	313·1	374	150	363·4	205
78	314·0	370	160	368·7	193
79	314·9	366	170	373·6	183
80	315·8	362	180	378·4	174
81	316·7	358	190	382·9	166
82	317·6	354	200	387·3	158
83	318·4	350	210	391·5	151
84	319·3	346	220	395·5	145

Explanation of the following Table.

The first column gives the absolute pressure of the steam in inches of mercury, or the height to which the pressure would raise a column of mercury in a tube, provided the opposing pressure of the atmosphere were removed.

The second column gives the absolute pressure in pounds per square inch under the same circumstances.

The third column, it will be observed, is headed "Pressure above Atmosphere." By this is meant the apparent pressure of the steam as indicated by a steam-gauge.

The fourth column shows the temperature in degrees of Fahrenheit's scale.

The fifth column shows the increase of volume which the water assumes in the act of changing into steam.

The sixth column shows the velocity with which steam, at the given pressures, escapes through an orifice into the atmosphere, as, for example, through the safety-valve of a steam-boiler.

TABLE

OF THE ELASTIC FORCE, TEMPERATURE, AND VOLUME OF STEAM FROM A TEMPERATURE OF 32° TO 457° FAH., AND FROM A PRESSURE OF 0·2 TO 900 INCHES OF MERCURY.

ELASTIC FORCE IN		Pressure above Atmosphere.	Temperature.	Volume.	Velocity of Escape.
Inches of Mercury.	Pounds per Square Inch.				
·200	·098		32°	187407	
·221	·108		35	170267	
·263	·129		40	144529	
·316	·155		45	121483	
·375	·184		50	103350	
·443	·217		55	88388	
·524	·257		60	75421	
·616	·302		65	64762	
·721	·353		70	55862	
·851	·417		75	47771	
1·000	·490		80	41031	

TABLE

OF THE ELASTIC FORCE, TEMPERATURE, AND VOLUME OF STEAM FROM A TEMPERATURE OF 32° TO 457° FAH., AND FROM A PRESSURE OF 0·2 TO 900 INCHES OF MERCURY.

ELASTIC FORCE IN		Pressure above Atmosphere.	Temperature.	Volume.	Velocity of Escape.
Inches of Mercury.	Pounds per Square Inch.				
1·17	·573		85·°	35393	
1·36	·666		90·	30425	
1·58	·774		95·	26686	
1·86	·911		100·	22873	
2·04	1·000		103·	20958	
2·18	1·068		105·	19693	
2·53	1·24		110·	16667	
2·92	1·431		115·	14942	
3·33	1·632		120·	13215	
3·79	1·857		125·	11723	
4·34	2·129		130·	10328	
5·00	2·45		135·	9036	
5·74	2·813		140·	7938	
6·53	3·100		145·	7040	
7·42	3·636		150·	6243	
8·40	4·116		155·	5559	
9·46	4·635		160·	4976	
10·68	5·23		165·	4443	
12·13	5·94		170·	3943	
13·62	6·67		175·	3538	
15·15	7·42		180·	3208	
17·00	8·33		185·	2879	
19·00	9·31		190·	2595	
21·22	10·40		195·	2342	
23·64	11·58		200·	2118	
26·13	12·80		205·	1932	
28·84	14·13		210·	1763	
29·41	14·41		211·	1730	
30·00	14·70	0·	212·	1700	
30·60	15·00		212·8	1669	
31·62	15·50	0·8	214·5	1618	
32·64	16·00	1·3	216·3	1573	
33·66	16·50		218·	1530	
34·68	17·00	2·3	219·6	1488	
35·70	17·50		221·2	1440	
36·72	18·00	3·3	222·7	1411	
37·74	18·50		224·2	1377	874
38·76	19·00	4·3	225·6	1343	

TABLE

OF THE ELASTIC FORCE, TEMPERATURE, AND VOLUME OF STEAM FROM A TEMPERATURE OF 32° TO 457° FAH., AND FROM A PRESSURE OF 0·2 TO 900 INCHES OF MERCURY.

ELASTIC FORCE IN		Pressure above Atmosphere.	Temper-ature.	Volume.	Velocity of Escape.
Inches of Mercury.	Pounds per Square Inch.				
39·78	19·50		227·1°	1312	
40·80	20·00	**5·3**	228·5	1281	
41·82	20·50		229·9	1253	
42·84	21·00	**6·3**	231·2	1225	
43·86	21·50		232·5	1199	
44·88	22·00	**7·3**	233·8	1174	1135
45·90	22·50		235·1	1150	
46·92	23·00	**8·3**	236·3	1127	
47·94	23·50		237·5	1105	
48·96	24·00	**9·3**	238·7	1084	
49·98	24·50		239·9	1064	
51·00	25·00	**10·3**	241·	1044	
53·04	26·00	**11·3**	243·3	1007	1295
55·08	27·	**12·3**	245·5	973	
57·12	28·	**13·3**	247·6	941	
59·16	29·	**14·3**	249·6	911	1407
61·20	30·	**15·3**	251·6	883	
63·24	31·	**16·3**	253·6	857	
65·28	32·	**17·3**	255·5	833	
67·32	33·	**18·3**	257·3	810	1491
69·36	34·	**19·3**	259·1	788	
71·40	35·	**20·3**	260·9	767	
73·44	36·	**21·3**	262·6	748	
75·48	37·	**22·3**	264·3	729	1550
77·52	38·	**23·3**	265·9	712	
79·56	39·	**24·3**	267·5	695	
81·60	40·	**25·3**	269·1	679	1600
83·64	41·	**26·3**	270·6	664	
85·68	42·	**27·3**	272·1	649	
87·72	43·	**28·3**	273·6	635	
89·76	44·	**29·3**	275·	622	1652
91·80	45·	**30·3**	276·4	610	
93·84	46·	**31·3**	277·8	598	
95·88	47·	**32·3**	279·2	586	
97·92	48·	**33·3**	280·5	575	1690
99·96	49·	**34·3**	281·9	564	
102·00	50·	**35·3**	283·2	554	
104·04	51·	**36·3**	284·4	544	1720

TABLE

OF THE ELASTIC FORCE, TEMPERATURE, AND VOLUME OF STEAM FROM A TEMPERATURE OF 32° TO 457° FAH., AND FROM A PRESSURE OF 0·2 TO 900 INCHES OF MERCURY.

ELASTIC FORCE IN		Pressure above Atmosphere.	Temperature.	Volume.	Velocity of Escape.
Inches of Mercury.	Pounds per Square Inch.				
106·08	52·	**37·3**	285·7°	534	
108·12	53·	**38·3**	286·9	525	
110·16	54·	**39·3**	288·1	516	
112·20	55·	**40·3**	289·3	508	1750
114·24	56·	**41·3**	290·5	500	
116·28	57·	**42·3**	291·7	492	
118·32	58·	**43·3**	292·9	484	1774
120·36	59·	**44·3**	294·2	477	
122·40	60·	**45·3**	295·6	470	
124·44	61·	**46·3**	296·9	463	
126·48	62·	**47·3**	298·1	456	
128·52	63·	**48·3**	299·2	449	
130·66	64·	**49·3**	300·3	443	
132·60	65·	**50·3**	301·3	437	
134·64	66·	**51·3**	302·4	431	1816
136·68	67·	**52·3**	303·4	425	
138·72	68·	**53·3**	304·4	419	
140·76	69·	**54·3**	305·4	414	
142·80	70·	**55·3**	306·4	408	
144·84	71·	**56·3**	307·4	403	
146·88	72·	**57·3**	308·4	398	
148·92	73·	**58·3**	309·3	393	1850
150·96	74·	**59·3**	310·3	388	
153·02	75·	**60·3**	311·2	383	
155·06	76·	**61·3**	312·2	379	
157·10	77·	**62·3**	313·1	374	
159·14	78·	**63·3**	314·	370	
161·18	79·	**64·3**	314·9	366	
163·22	80·	**65·3**	315·8	362	
165·26	81·	**66·3**	316·7	358	
167·30	82·	**67·3**	317·7	354	
169·34	83·	**68·3**	318·4	350	
171·38	84·	**69·3**	319·3	346	
173·42	85·	**70·3**	320·1	342	
183·62	90·	**75·3**	324·3	325	1904
193·82	95·	**80·3**	328·2	310	
203·99	100·	**85·3**	332·	295	
214·19	105·	**90·3**	335·8	282	1950

TABLE

OF THE ELASTIC FORCE, TEMPERATURE, AND VOLUME OF STEAM FROM A TEMPERATURE OF 32° TO 457° FAH., AND FROM A PRESSURE OF 0·2 TO 900 INCHES OF MERCURY.

ELASTIC FORCE IN		Pressure above Atmosphere.	Temperature.	Volume.	Velocity of Escape.
Inches of Mercury.	Pounds per Square Inch.				
224·39	110·	**95·3**	339·2°	271	
234·59	115·	**100·3**	342·7	259	
244·79	120·	**105·3**	345·8	251	1980
254·99	125·	**110·3**	349·1	240	
265·19	130·	**115·3**	352·1	233	
275·39	135·	**120·3**	355·	224	2006
285·59	140·	**125·3**	357·9	218	
295·79	145·	**130·3**	360·6	210	
306·	150·	**135·3**	363·4	205	2029
316·19	155·	**140·3**	366·	198	
326·29	160·	**145·3**	368·7	193	
336·59	165·	**150·3**	371·1	187	
346·79	170·	**155·3**	373·6	183	
357·	175·	**160·3**	376·	178	
367·2	180·	**165·3**	378·4	174	
377·1	185·	**170·3**	380·6	169	2074
387·6	190·	**175·3**	382·9	166	
397·8	195·	**180·3**	384·1	161	
408·	200·	**185·3**	387·3	158	
448·8	220·	**205·3**	392·		2109
524·28	257·	**242·3**	406·		2136
599·76	294·	**279·3**	418·		2159
848·68	367·	**352·3**	429·		2196
889·64	441·	**426·3**	457·		2226

It will be observed that in the foregoing and following tables the relative volume and weight of steam differs with different authors, and, while they may not all be scientifically correct, they are undoubtedly approximately so, or sufficiently correct for all practical purposes. Therefore, it would be perfectly safe to take the volume of steam at 1728; in other words, a cubic inch of water converted into steam at atmospheric pressure will occupy 1728 cubic inches, or one cubic foot.

TABLE

SHOWING THE TEMPERATURE AND WEIGHT OF STEAM AT DIFFERENT PRESSURES FROM 1 POUND PER SQUARE INCH TO 300 POUNDS, AND THE QUANTITY OF STEAM PRODUCED FROM 1 CUBIC INCH OF WATER, ACCORDING TO PRESSURE.

Total Pressure per Square Inch measured from a Vacuum.	Pressure above Atmosphere.	Sensible Temperature in Fahrenheit degrees.	Total Heat in Degrees from Zero of Fahrenheit.	Weight of one Cubic Foot of Steam.	Relative Volume of Steam compared with Water from which it was raised.
1	......	102·1	1144·5	·0030	20582
2	......	126·3	1151·7	·0058	10721
3	......	141·6	1156·6	·0085	7322
4	......	153·1	1160·1	·0112	5583
5	......	162·3	1162·9	·0138	4527
6	......	170·2	1165·3	·0163	3813
7	......	176·9	1167·3	·0189	3298
8	......	182·9	1169·2	·0214	2909
9	......	188·3	1170·8	·0239	2604
10	......	193·3	1172·3	·0264	2358
11	......	197·8	1173·7	·0289	2157
12	......	202·0	1175·0	·0314	1986
13	......	205·9	1176·2	·0338	1842
14	......	209·6	1177·3	·0362	1720
14·7	0	212·0	1178·1	·0380	1642
15	·3	213·1	1178·4	·0387	1610
16	1·3	216·3	1179·4	·0411	1515
17	2·3	219·6	1180·3	·0435	1431
18	3·3	222·4	1181·2	·0459	1357
19	4·3	225·3	1182·1	·0483	1290
20	5·3	228·0	1182·9	·0507	1229
21	6·3	230·6	1183·7	·0531	1174
22	7·3	233·1	1184·5	·0555	1123
23	8·3	235·5	1185·2	·0580	1075
24	9·3	237·8	1185·9	·0601	1036
25	10·3	240·1	1186·6	·0625	996
26	11·3	242·3	1187·3	·0650	958
27	12·3	244·4	1187·8	·0673	926
28	13·3	246·4	1188·4	·0696	895
29	14·3	248·4	1189·1	·0719	866
30	15·3	250·4	1189·8	·0743	838
31	16·3	252·2	1190·4	·0766	813
32	17·3	254·1	1190·9	·0789	789
33	18·3	255·9	1191·5	·0812	767
34	19·3	257·6	1192·0	·0835	746
35	20·3	259·3	1192·5	·0858	726
36	21·3	260·9	1193·0	·0881	707
37	22·3	262·6	1193·5	·0905	688
38	23·3	264·2	1194·0	·0929	671
39	24·3	265·8	1194·5	·0952	655

TABLE — (*Continued.*)

Total Pressure per Square Inch measured from a Vacuum.	Pressure above Atmosphere.	Sensible Temperature in Fahrenheit degrees.	Total Heat in Degrees from Zero of Fahrenheit.	Weight of one Cubic Foot of Steam.	Relative Volume of Steam compared with Water from which it was raised.
40	25·3	267·3	1194·9	·0974	640
41	26·3	268·7	1195·4	·0996	625
42	27·3	270·2	1195·8	·1020	611
43	28·3	271·6	1196·2	·1042	598
44	29·3	273·0	1196·6	·1065	595
45	30·3	274·4	1197·1	·1089	572
46	31·3	275·8	1197·5	·1111	561
47	32·3	277·1	1197·9	·1133	550
48	33·3	278·4	1198·3	·1156	539
49	34·3	279·7	1198·7	·1179	529
50	35·3	281·0	1199·1	·1202	518
51	36·3	282·3	1199·5	·1224	509
52	37·3	283·5	1199·9	·1246	500
53	38·3	284·7	1200·3	·1269	491
54	39·3	285·9	1200·6	·1291	482
55	40·3	287·1	1201·0	·1314	474
56	41·3	288·2	1201·3	·1336	466
57	42·3	289·3	1201·7	·1364	458
58	43·3	290·4	1202·0	·1380	451
59	44·3	291·6	1202·4	·1403	444
60	45·3	292·7	1202·7	·1425	437
61	46·3	293·8	1203·1	·1447	430
62	47·3	294·8	1203·4	·1469	424
63	48·3	295·9	1203·7	·1493	417
64	49·3	296·9	1204·0	·1516	411
65	50·3	298·0	1204·3	·1538	405
66	51·3	299·0	1204·6	·1560	399
67	52·3	300·0	1204·9	·1583	393
68	53·3	300·9	1205·2	·1605	388
69	54·3	301·9	1205·5	·1627	383
70	55·3	302·9	1205·8	·1648	378
71	56·3	303·9	1206·1	·1670	373
72	57·3	304·8	1206·3	·1692	368
73	58·3	305·7	1206·6	·1714	363
74	59·3	306·6	1206·9	·1736	359
75	60·3	307·5	1207·2	·1759	353
76	61·3	308·4	1207·4	·1782	349
77	62·3	309·3	1207·7	·1804	345
78	63·3	310·2	1208·0	·1826	341
79	64·3	311·1	1208·3	·1848	337
80	65·3	312·0	1208·5	·1869	333
81	66·3	312·8	1208·8	·1891	329
82	67·3	313·6	1209·1	·1913	325
83	68·3	314·5	1209·4	·1935	321

TABLE — (*Continued.*)

Total Pressure per Square Inch measured from a Vacuum.	Pressure above Atmosphere.	Sensible Temperature in Fahrenheit degrees.	Total Heat in Degrees from Zero of Fahrenheit.	Weight of one Cubic Foot of Steam.	Relative Volume of Steam compared with Water from which it was raised.
84	69·3	315·3	1209·6	·1957	318
85	70·3	316·1	1209·9	·1980	314
86	71·3	316·9	1210·1	·2002	311
87	72·3	317·8	1210·4	·2024	308
88	73·3	318·6	1210·6	·2044	305
89	74·3	319·4	1210·9	·2067	301
90	75·3	320·2	1211·1	·2089	298
91	76·3	321·0	1211·3	·2111	295
92	77·3	321·7	1211·5	·2133	292
93	78·3	322·5	1211·8	·2155	289
94	79·3	323·3	1212·0	·2176	286
95	80·3	324·1	1212·3	·2198	283
96	81·3	324·8	1212·5	·2219	281
97	82·3	325·6	1212·8	·2241	278
98	83·3	326·3	1213·0	·2263	275
99	84·3	327·1	1213·2	·2285	272
100	85·3	327·9	1213·4	·2307	270
101	86·3	328·5	1213·6	·2329	267
102	87·3	329·1	1213·8	·2351	265
103	88·3	329·9	1214·0	·2373	262
104	89·3	330·6	1214·2	·2393	260
105	90·3	331·3	1214·4	·2414	257
106	91·3	331·9	1214·6	·2435	255
107	92·3	332·6	1214·8	·2456	253
108	93·3	333·3	1215·0	·2477	251
109	94·3	334·0	1215·3	·2499	249
110	95·3	334·6	1215·5	·2521	247
111	96·3	335·3	1215·7	·2543	245
112	97·3	336·0	1215·9	·2564	243
113	98·3	336·7	1216·1	·2586	241
114	99·3	337·4	1216·3	·2607	239
115	100·3	338·0	1216·5	·2628	237
116	101·3	338·6	1216·7	·2649	235
117	102·3	339·3	1216·9	·2674	233
118	103·3	339·9	1217·1	·2696	231
119	104·3	340·5	1217·3	·2738	229
120	105·3	341·1	1217·4	·2759	227
121	106·3	341·8	1217·6	·2780	225
122	107·3	342·4	1217·8	·2801	224
123	108·3	343·0	1218·0	·2822	222
124	109·3	343·6	1218·2	·2845	221
125	110·3	344·2	1218·4	·2867	219
126	111·3	344·8	1218·6	·2889	217
127	112·3	345·4	1218·8	·2911	215

TABLE — (*Concluded.*)

Total Pressure per Square Inch measured from a Vacuum.	Pressure above Atmosphere.	Sensible Temperature in Fahrenheit degrees.	Total Heat in Degrees from Zero of Fahrenheit.	Weight of one Cubic Foot of Steam.	Relative Volume of Steam compared with Water from which it was raised.
128	113·3	346·0	1218·9	·2933	214
129	114·3	346·6	1219·1	·2955	212
130	115·3	347·2	1219·3	·2977	211
131	116·3	347·8	1219·5	·2999	209
132	117·3	348·3	1219·6	·3020	208
133	118·3	348·9	1219·8	·3040	206
134	119·3	349·5	1220·0	·3060	205
135	120·3	350·1	1220·2	·3080	203
136	121·3	350·6	1220·3	·3101	202
137	122·3	351·2	1220·5	·3121	200
138	123·3	351·8	1220·7	·3142	199
139	124·3	352·4	1220·9	·3162	198
140	125·3	352·9	1221·0	·3184	197
141	126·3	353·5	1221·2	·3206	195
142	127·3	354·0	1221·4	·3228	194
143	128·3	354·5	1221·6	·3258	193
144	129·3	355·0	1221·7	·3273	192
145	130·3	355·6	1221·9	·3294	190
146	131·3	356·1	1222·0	·3315	189
147	132·3	356·7	1222·2	·3336	188
148	133·3	357·2	1222·3	·3357	187
149	134·3	357·8	1222·5	·3377	186
150	135·3	358·3	1222·7	·3397	184
155	140·3	361·0	1223·5	·3500	179
160	145·3	363·4	1224·2	·3607	174
165	150·3	366·0	1224·9	·3714	169
170	155·3	368·2	1225·7	·3821	164
175	160·3	370·8	1226·4	·3928	159
180	165·3	372·9	1227·1	·4035	155
185	170·3	375·3	1227·8	·4142	151
190	175·3	377·5	1228·5	·4250	148
195	180·3	379·7	1229·2	·4357	144
200	185·3	381·7	1229·8	·4464	141
210	195·3	386·0	1231·1	·4668	135
220	205·3	389·9	1232·8	·4872	129
230	215·3	393·8	1233·5	·5072	123
240	225·3	397·5	1234·6	·5270	119
250	235·3	401·1	1235·7	·5471	114
260	245·3	404·5	1236·8	·5670	110
270	255·3	407·9	1237·8	·5871	106
280	265·3	411·2	1238·8	·6070	102
290	275·3	414·4	1239·8	·6268	99
300	285·3	417·5	1240·7	·6469	96

TABLE

SHOWING THE STEAM PRESSURE IN POUNDS PER GAUGE; THE ABSOLUTE PRESSURE IN POUNDS AND INCHES OF MERCURY; THE TEMPERATURE; THE TOTAL HEAT IN STEAM PER POUND; THE LATENT HEAT PER POUND; THE HEAT OF THE WATER; THE RELATIVE VOLUME, AND WEIGHT OF STEAM PER CUBIC FOOT, FOR VARIOUS PRESSURES.*

Pressure per Gauge.	Total lbs.	Inches of Mercury.	Temperature, Fah.	Total Heat per lb.	Latent Heat per lb.	Heat in Water per lb.	Relative Volume.	Weight per Cub. Ft.
	1	2·036	102·	1145·05	1042·96	102·08	17983·	·00347
	2	4·072	126·27	1152·45	1026·01	126·44	10353·	·00602
	3	6·108	141·62	1157·13	1015·25	141·87	7283·8	·00856
	4	8·144	153·07	1162·62	1007·23	153·39	5608·4	·01112
	5	10·180	162·33	1163·45	1000·73	162·72	4565·6	·01366
	6	12·216	170·12	1165·83	995·25	170·57	3851·0	·01619
	7	14·252	176·91	1167·89	990·47	177·42	3330·8	·01837
	8	16·288	182·91	1169·72	986·24	183·48	2935·1	·02125
	9	18·324	188·32	1171·37	982·43	188·94	2624·1	·02377
	10	20·360	193·24	1172·87	978·96	193·92	2373·0	·02628
	11	22·396	197·77	1174·26	975·76	198·49	2166·3	·02880
	12	24·432	201·96	1175·53	972·80	202·74	1993·0	·03130
	13	26·468	205·88	1176·73	970·02	206·71	1845·7	·03380
	14	28·504	209·56	1177·85	967·43	210·43	1718·9	·03629
·304	15	30·540	213·02	1178·91	964·97	213·94	1608·6	·03878
1·304	16	32·576	216·30	1179·91	962·66	217·25	1511·7	·04123
2·304	17	34·612	219·41	1180·86	960·45	220·41	1426·2	·04374
3·304	18	36·648	222·38	1181·76	958·34	223·42	1349·8	·04622
4·304	19	38·684	225·20	1182·63	956·34	226·28	1281·1	·04868
5·304	20	40·720	227·92	1183·45	954·41	229·04	1219·7	·05119
6·304	21	42·756	230·51	1184·25	952·57	231·67	1163·8	·05360
7·304	22	44·792	233·02	1185·01	950·79	234·22	1112·9	·05605
8·304	23	46·828	235·43	1185·74	949·07	236·67	1066·3	·05851
9·304	24	48·864	237·75	1186·45	947·42	239·03	1023·6	·06095
10·304	25	50·900	240·00	1187·14	945·82	241·31	984·23	·06338
11·304	26	52·936	242·17	1187·80	944·28	243·52	947·86	·06582
12·304	27	54·972	244·28	1188·44	942·77	245·67	914·14	·06824
13·304	28	57·008	246·33	1189·07	941·32	247·75	882·80	·07067
14·304	29	59·044	248·31	1189·67	939·90	249·77	853·60	·07308
15·304	30	61·080	250·24	1190·26	938·92	251·74	826·32	·07550
16·304	31	63·116	252·12	1190·83	937·19	253·64	800·79	·07791
17·304	32	65·152	253·95	1191·40	935·88	255·52	766·83	·08031
18·304	33	67·188	255·73	1191·94	934·61	257·33	754·31	·08271
19·304	34	69·224	257·46	1192·47	933·36	259·11	733·09	·08510
20·304	35	71·260	259·17	1192·99	932·15	260·84	713·08	·08749
21·304	36	73·296	260·83	1193·49	930·96	262·53	694·17	·08987
22·304	37	75·331	262·46	1193·99	929·81	264·18	676·27	·09225
23·304	38	77·367	264·04	1194·47	928·67	265·80	659·31	·09462
24·304	39	79·403	265·60	1194·94	927·56	267·38	643·21	·09700
25·304	40	81·439	267·12	1195·41	926·47	268·94	627·91	·09936
26·304	41	83·475	268·61	1195·86	925·40	270·46	613·34	·10172
27·304	42	85·511	270·07	1196·31	924·36	271·95	599·46	·10407
28·304	43	87·547	271·51	1196·75	923·33	273·42	586·23	·10642
29·304	44	89·583	272·91	1197·18	922·32	274·86	573·58	·10877
30·304	45	91·619	274·29	1197·60	921·33	276·27	561·50	·11111
31·304	46	93·655	275·65	1198·01	920·36	277·65	549·94	·11344
32·304	47	95·691	276·99	1198·42	919·40	279·02	538·87	·11577
33·304	48	97·727	278·30	1198·82	918·47	280·35	528·25	·11810
34·304	49	99·763	279·58	1199·21	917·54	281·67	518·07	·12042
35·304	50	101·799	280·85	1199·60	916·63	282·97	508·29	·12273

* John W. Hill.

TABLE

SHOWING THE STEAM PRESSURE IN POUNDS PER GAUGE; THE ABSOLUTE PRESSURE IN POUNDS AND INCHES OF MERCURY; THE TEMPERATURE; THE TOTAL HEAT IN STEAM PER POUND; THE LATENT HEAT PER POUND; THE HEAT OF THE WATER; THE RELATIVE VOLUME, AND WEIGHT OF STEAM PER CUBIC FOOT, FOR VARIOUS PRESSURES.

Pressure per Gauge.	Total lbs.	Inches of Mercury.	Temperature, Fah.	Total Heat per lb.	Latent Heat per lb.	Heat in Water per lb.	Relative Volume.	Weight per Cub. Ft.
36·304	51	103·84	282·10	1198·98	915·74	284·24	498·89	·12505
37·304	52	105·87	283·32	1200·35	914·86	285·50	489·85	·12736
38·304	53	107·91	284·53	1200·72	913·99	286·73	481·15	·12966
39·304	54	109·94	285·72	1201·08	913·13	287·95	472·77	·13196
40·304	55	111·98	286·89	1201·44	912·29	289·15	464·69	·13428
41·304	56	114·02	288·05	1201·80	911·46	290·34	456·90	·13652
42·304	57	116·05	289·11	1202·14	910·64	291·50	449·38	·13883
43·304	58	118·09	290·32	1202·49	909·83	292·65	442·12	·14111
44·304	59	120·12	291·42	1202·82	909·03	293·79	435·10	·14338
45·304	60	122·16	292·52	1203·16	908·25	294·91	428·32	·14566
46·304	61	124·19	293·60	1203·49	907·47	296·02	421·75	·14792
47·304	62	126·23	294·66	1203·81	906·70	297·11	415·40	·15018
48·304	63	128·27	295·71	1204·13	905·95	298·18	409·25	·15244
49·304	64	130·30	296·75	1204·45	905·20	299·25	403·29	·15469
50·304	65	132·34	297·78	1204·76	904·46	300·30	397·51	·15694
51·304	66	134·37	298·79	1205·07	903·73	301·34	391·90	·15919
52·304	67	136·41	299·79	1205·38	903·01	302·37	386·47	·16130
53·304	68	138·45	300·77	1205·68	902·30	303·38	381·18	·16366
54·304	69	140·48	301·75	1205·97	901·60	304·37	376·06	·16590
55·304	70	142·52	302·72	1206·27	900·90	305·37	371·07	·16812
56·304	71	144·55	303·67	1206·56	900·21	306·35	366·24	·17035
57·304	72	146·59	304·62	1206·85	899·53	307·32	361·53	·17256
58·304	73	148·63	305·55	1207·13	898·85	308·28	356·95	·17478
59·304	74	150·66	306·47	1207·42	898·19	309·23	352·49	·17690
60·304	75	152·70	307·39	1207·69	897·53	310·16	348·15	·17919
61·304	76	154·73	308·29	1207·97	896·88	311·09	343·93	·18139
62·304	77	156·77	309·18	1208·24	896·23	312·01	339·81	·18359
63·304	78	158·81	310·07	1208·51	895·59	312·92	335·81	·18578
64·304	79	160·84	310·94	1208·78	894·95	313·82	331·89	·18797
65·304	80	162·88	311·81	1209·04	894·33	314·71	328·08	·19015
66·304	81	164·91	312·67	1209·30	893·71	315·59	324·37	·19233
67·304	82	166·95	313·52	1209·56	893·09	316·47	320·74	·19451
68·304	83	168·99	314·36	1209·82	892·49	317·33	317·20	·19668
69·304	84	171·02	315·19	1210·07	891·88	318·19	313·74	·19885
70·304	85	173·06	316·02	1210·33	891·29	319·04	310·36	·20101
71·304	86	175·09	316·84	1210·58	890·69	319·89	307·07	·20317
72·304	87	177·13	317·65	1210·83	890·11	320·72	303·85	·20532
73·304	88	179·17	318·45	1211·07	889·52	321·54	300·70	·20747
74·304	89	181·20	319·25	1211·31	888·95	322·36	297·62	·20962
75·304	90	185·24	320·04	1211·55	888·38	323·17	294·61	·21185
76·304	91	185·27	320·82	1211·79	887·81	323·98	291·66	·21390
77·304	92	187·31	321·58	1212·03	887·25	324·78	288·78	·21603
78·304	93	189·35	322·36	1212·26	886·69	325·57	285·96	·21816
79·304	94	191·38	323·13	1212·49	886·13	326·36	283·21	·22029
80·304	95	193·42	323·88	1212·72	885·59	327·13	280·50	·22241
81·304	96	195·45	324·63	1212·95	885·04	327·91	277·86	·22453
82·304	97	197·49	325·38	1213·18	884·50	328·68	275·27	·22675
83·304	98	199·53	326·11	1213·40	883·97	329·43	272·73	·22873
84·304	99	201·56	326·84	1213·63	883·44	330·19	270·24	·23085
85·304	100	203·60	327·57	1213·85	882·91	330·94	267·80	·23296

TABLE

SHOWING THE STEAM PRESSURE IN POUNDS PER GAUGE; THE ABSOLUTE PRESSURE IN POUNDS AND INCHES OF MERCURY; THE TEMPERATURE; THE TOTAL HEAT IN STEAM PER POUND; THE LATENT HEAT PER POUND; THE HEAT OF THE WATER; THE RELATIVE VOLUME AND WEIGHT OF STEAM PER CUBIC FOOT, FOR VARIOUS PRESSURES.

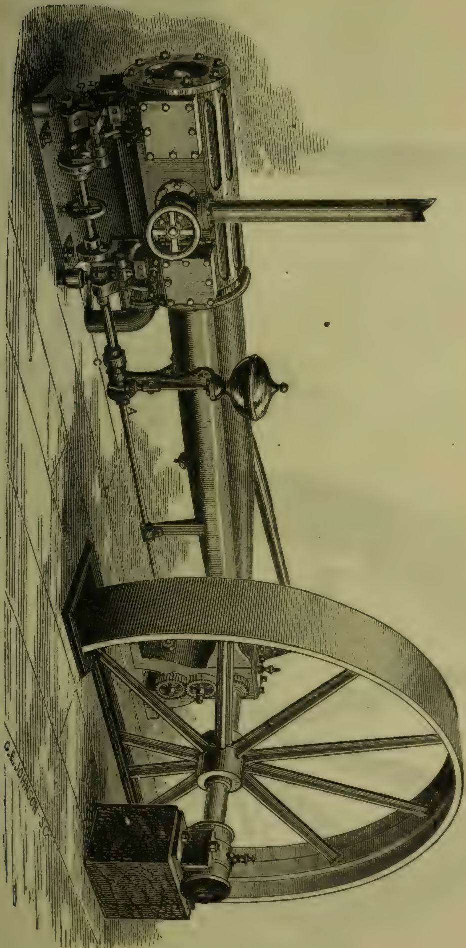
Pressure per Gauge.	Total lbs.	Inches of Mercury.	Temperature, Fah.	Total Heat per lb.	Latent Heat per lb.	Heat in Water per lb.	Relative Volume.	Weight per Cub. Ft.
86·304	101	205·64	328·29	1214·07	882·39	331·68	265·81	·23505
87·304	102	207·67	329·00	1214·28	881·87	332·41	263·07	·23715
88·304	103	209·71	329·71	1214·50	881·35	333·15	260·77	·23924
89·304	104	211·74	330·42	1214·71	880·85	333·86	258·52	·24132
90·304	105	213·78	331·11	1214·93	880·34	334·59	256·31	·24340
91·304	106	215·82	331·80	1215·14	879·84	335·30	254·14	·24548
92·304	107	217·85	332·49	1215·35	879·34	336·01	252·01	·24756
93·304	108	219·89	333·17	1215·55	878·84	336·71	249·92	·24963
94·304	109	221·92	333·85	1215·76	878·35	337·41	247·87	·25169
95·304	110	223·96	334·52	1215·97	877·86	338·11	245·86	·25375
96·304	111	225·99	335·19	1216·17	877·38	338·79	243·88	·25581
97·304	112	228·03	335·85	1216·38	876·90	339·48	241·94	·25786
98·304	113	230·07	336·51	1216·58	876 42	340·16	240·03	·25991
99·304	114	232·10	337·16	1216·77	875·94	340·83	238·15	·26204
100·304	115	234·14	337·81	1216·97	875·47	341·50	236·31	·26400
101·304	116	236·17	338·46	1217·17	875·00	342·17	234·50	·26611
102·304	117	238·21	339·10	1217·36	874·54	342·83	232·70	·26816
103·304	118	240·25	339·73	1217·56	874·07	343·49	231·00	·27020
104·304	119	242·28	340·37	1217·75	873·61	344·14	229·30	·27224
105·304	120	244·32	340·99	1217·94	873·15	344·79	227·56	·27421
106·304	121	246·35	341·62	1218·13	872·70	345·43	226·00	·27628
107·304	122	248·39	342·24	1218·32	872·25	346·07	224·40	·27828
108·304	123	250·43	342·85	1218·51	871·80	346·71	222·80	·28027
109·304	124	252·46	343·46	1218·69	871·35	347·34	221·20	·28227
110·304	125	254·50	344·07	1218·88	870·91	347·97	219·50	·28422
111·304	126	256·54	344·68	1219·07	870·47	348·60	218·20	·28625
112·304	127	258·57	345·28	1219·25	870·03	349·22	216·70	·28824
113·304	128	260·61	345·87	1219·43	869·60	349·83	215·20	·29023
114·304	129	262 64	346·46	1219·61	869·16	350·45	213·70	·29222
115·304	130	264·68	347·06	1219·79	868·74	351·06	212·07	·29419
116·304	131	266·72	347·64	1219·97	868·31	351·66	210·90	·29618
117·304	132	268·75	348·23	1220·15	867·88	352·27	209·50	·29816
118·304	133	270·79	348·80	1220·32	867·46	352·86	208·10	·30013
119·304	134	272·82	349·38	1220·50	867·04	353·46	206·70	·30209
120·304	135	274·86	349·95	1220·67	866·62	354·05	205·18	·30406
121·304	136	276·89	350·52	1220·85	866·21	354·64	204·10	·30601
122·304	137	278·93	351·09	1221·02	865·79	355·23	202·80	·30796
123·304	138	280·96	351·75	1221·19	865·38	355·81	201·50	·30990
124·304	139	283·00	352·21	1221·36	864·97	356·39	200·20	·31186
125·304	140	285·04	352·76	1221·53	864·56	356·97	198·78	·31385
126·304	141	287·07	353·32	1221·70	864·16	357·54	197·80	·31586
127·304	142	289·11	353·87	1221·87	863·76	358·11	196·60	·31788
128·304	143	291·15	354·42	1222·04	863·36	358·67	195·40	·31990
129·304	144	293·18	354·96	1222·20	862·96	359·24	194·20	·32190
130·304	145	295·22	355·50	1222·37	862·57	359·80	192·83	·32354
131·304	146	297·25	356·04	1222·53	862·17	360·36	191·90	·32592
132·304	147	299·29	356·57	1222·69	861·78	360·91	190·80	·32794
133·304	148	301·33	357·10	1222·85	861·39	361·46	189·70	·32995
134·304	149	303·36	357·63	1223·02	861·01	362·01	188·60	·33196
135·304	150	305·40	358·16	1223·18	860·62	362·56	187·26	·33315

The Brown Automatic Cut-Off Steam-Engine.

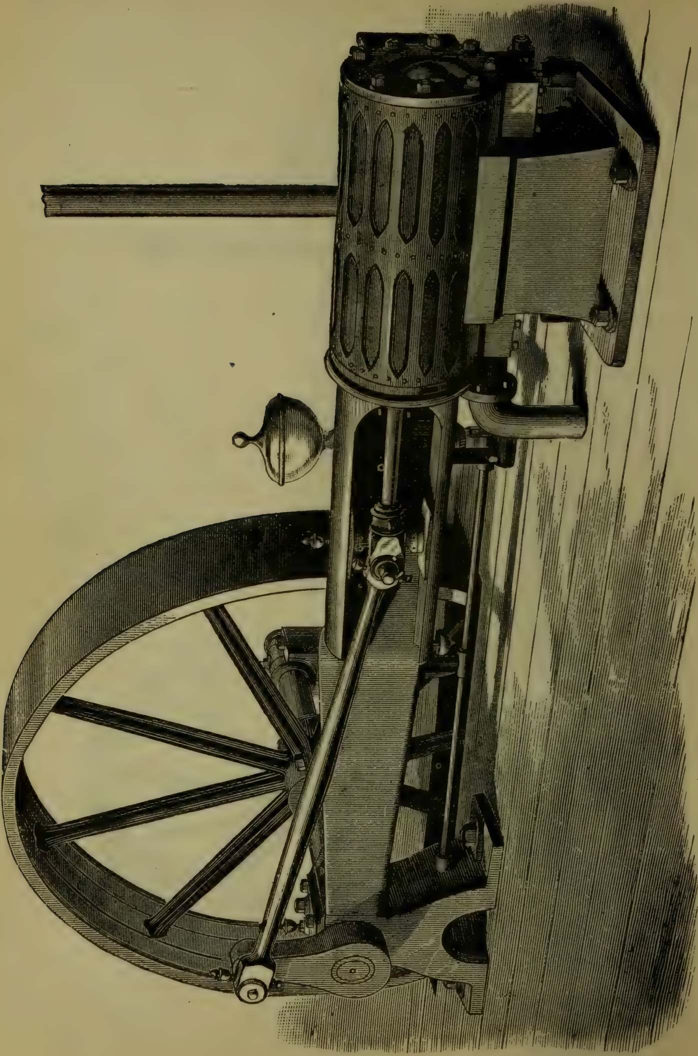
The cuts on pages 89, 90, represent the Brown Automatic Cut-off High-pressure Engine.—The housing, which, as will be observed, is of the girder-frame pattern, somewhat resembles the Corliss, though the engine is different in every other respect. The cylinder, which contains the steam- and exhaust-ports, is encased in an ornamental cast-iron jacket, and rests on a square, tapering column which extends nearly the full length. By a judicious distribution of the materials, every part possesses sufficient rigidity, without extra weight of metal. In its design, the evils induced by expansion, and the liability to get out of line, have been scientifically considered and practically obviated. A spur-gear on the main shaft gives motion to a shaft parallel with and below the axis of the cylinder. From this shaft the motions of the valves are derived.

There are four valves, one steam and one exhaust at each end of the cylinder, which are independent, and though slide-valves, as they have but one function to perform for each revolution, *i. e.*, admitting or exhausting the steam, they are necessarily of a different construction from the ordinary slide-valve. The exhaust-valves are horizontal, and travel at right angles with the cylinder; the motion being derived from cams on the longitudinal shaft, which is positive in both directions. The shape of the camways is such that the motions of the valves in opening and closing are very quick, and allow of their remaining stationary during nearly the whole stroke of the piston, thus insuring a perfectly free exhaust, and preventing any possibility of back pressure.

The steam-valves, which are vertical, are of the gridiron pattern, and require very little movement to give a full port opening. They are operated by eccentrics on the cam-shaft, in connection with the following device for regulating the point of cut-off. A vibrating lifter, having the fixed centre at its outer end, is connected, at about the middle of its length, with the eccentric-rod; while the inner end engages a spring-catch or projection on the



The Brown Automatic Cut-Off Engine.



Front View of the Brown Automatic Cut-Off Engine.

valve-stem, giving to the valve a positive motion on the left or up stroke, and allowing of its being tripped, or released for closing, when the point of cut-off is reached—jar being prevented by means of small dash-pots. On the spring-catch of the valve-stems is an inverted wedge, by means of which the valves are tripped.

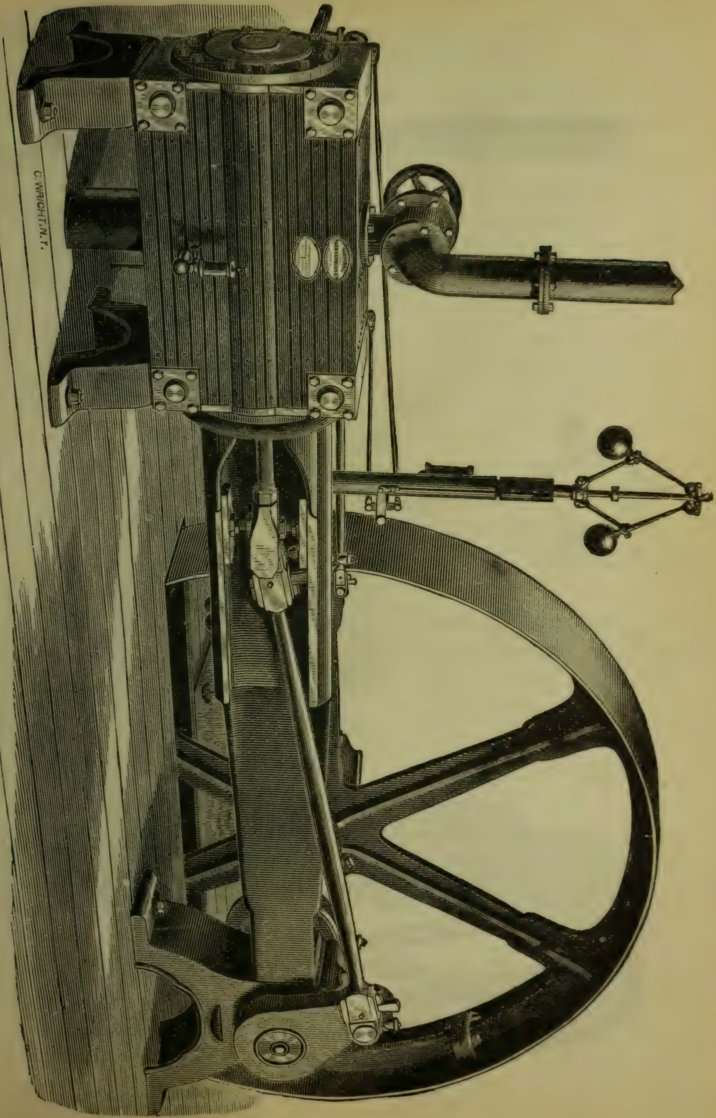
The governor, which, as will be observed, is enclosed in an ornamental case or shell, is very sensitive and admirably adapted to these engines, is of the centrifugal fly-ball type, receives a positive motion from the cam-shaft, by means of bevel-gears, and causes a rod running parallel with the shaft and back of the valve-stems to oscillate. On this rod and opposite to each wedge is an arm, which, when the speed increases, is moved by the governor towards the wedge, thus drawing the catch away from the lifter as it rises, and allowing the valve to drop, while the lifter continues its movment to the end of the stroke and return, when it engages the catch as before.

Both steam- and exhaust-valves have ample openings, which, in connection with their quick motions, entirely obviate the evil arising from wire-drawing the steam, or choking the exhaust, thereby causing back pressure. The only unbalanced pressure on the valve is an area of about one square inch on the top, for the purpose of aiding in closing them quickly. As an evidence of the small amount of friction induced in the working of the valves of these machines, the ordinary starting-bar is dispensed with, and an eight-inch hand-wheel on the cam-shaft, which possesses sufficient leverage to work the valve by hand, substituted in its place.

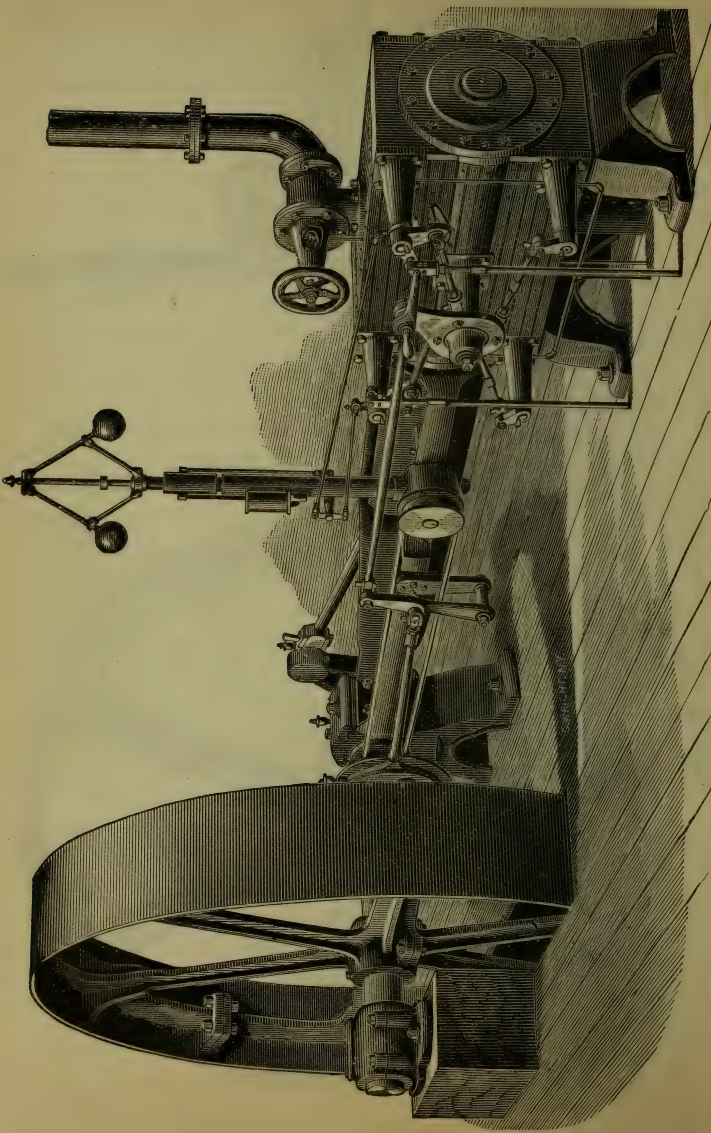
The valve-gear is a most ingenious and admirable combination of mechanical devices, being very simple, and susceptible of easy, convenient, and accurate adjustment. Its operation may be explained as follows: The shaft, *A*, receives its motion from a gear on the main shaft, which, in turn, imparts motion to the governor, and through the medium of the frictional device, or coupling *C*, to the shaft *B*, on which the eccentrics, *D*, are located, the ends of the straps of which are connected to the horizontal

arms, *E*, which extend into the square slot provided in the slide-spindle, and to the catch of the tongue. As the shaft, *B*, revolves, the ends of the arms, *E*, will reciprocate vertically in the square slot, the valve-stem being attached to the guide, *F*, in the slot of which the tongue, *G*, is pivoted by the pin shown in the guide. The upper end of this tongue has a projecting catch upon it, beneath which stands the end of the arm, *E*, which lifts the valve for the admission of the steam, and holds it open until the tongue is tripped, when the valve closes, the movement being instantaneous, and rendered noiseless by means of an air-cushion dash-pot.

The governor-spindle is attached to the end of an arm which is fast upon a rod, which, being immediately back of the shaft *B*, is not seen in the cut; upon this rod, and immediately behind the steam-valve spindle-guide, *F*, is an arm standing vertically, and carrying the horizontal pin, *H*. The tongue which at one end acts as a catch to the eccentric arm, at the other end protrudes from the back of the slide-spindle guide, and stands directly beneath the pin, *H*, so that when the arm, *E*, lifts (through the tongue-catch) the steam-valve, the latter remains open until the tail of the catch, *G*, meets with the pin, *H*, which trips the tongue and closes the valve. The governor controls the position of the pin, *H*, and determines the point of cut-off. The discs, *J*, on the shaft, *B*, are provided with cam-grooves, into which a friction-roller on the rocker-arm, *K*, extends, the upper arm, *L*, being attached to the exhaust-valve spindle. To compensate for the circular motion of the arm and the vertical movement of the valve-spindle, the connection between them is made by the eye of the spindle, containing a slot in which is fitted a sliding die, to which the pin of the arm is fitted. Any change of load on the engine is instantaneously shown by the governor. Nevertheless, the valve-gear is complicated, and liable to wear out rapidly and become a source of annoyance and expense.



Front View of the Harris Corliss Engine.



Back View of the Harris Corliss Engine.

The Harris Corliss Steam-Engine.

The cuts on pages 93 and 94 represent the Harris Corliss Engine, one showing the crank and cross-head, the other the eccentric and valve gear. It will be observed that the general design is symmetrical and well proportioned, rigidity and strength being introduced principally where the greatest longitudinal strain occurs, viz., between the cylinder-flange and the centre of the fly-wheel shaft. Between these points the frame, which is in one casting, is vertically deep and strongly ribbed, thus insuring greater strength and stiffness than could be obtained by any other distribution of the same amount of material. The cross-head guides are cast with the frame. The main pillow-block is of an improved design, with the feet well spread out; and the cylinder and exhaust-chest rest upon supports the entire width of the chest. The engines are only slightly elevated above the floor, thus allowing the attendant to reach every part with the greatest ease. The cylinders and chests are neatly lagged in black walnut, or other wood.

The piston-packing is of the most improved kind, and is claimed to remain perfectly steam-tight under all circumstances. It is set out by means of German silver spiral springs, which obviate the difficulty arising from the cylinder becoming worn larger at the ends, or its liability to become cut or fluted, in consequence of its being set out too tight. Besides, the piston-rod is retained exactly in the centre of the cylinder. The spring plates for the packing-rings are made of bronze metal, consequently they are not liable to corrode. The distance used for the packing-ring between the piston and the follower is so small that it leaves a large amount of the junk-ring for a bearing, or a wearing surface on the lower side of the cylinder in the horizontal engine, which reduces the liability to cut. The design and arrangement of this packing afford the most convenient facilities for taking it out, putting it in, or for adjustment. The operation of the packing is as follows: When steam is admitted into either end of the cylinder, the

packing-ring is carried by the steam over to the side of the groove in the junk-ring, making a joint there, and allowing the steam to pass down and under the packing-ring, thus placing it in equilibrium; then all that is required is a very light spring to hold the packing in contact with the cylinder.

There are four valves—two steam and two exhaust. The steam-valves are located on the top of the cylinder at each end, and open directly into the clearance, which obviates the waste induced by the use of long passages. The exhaust-valves are placed in the exhaust-chest on the lower side of the cylinder, and, as in the case of the steam-valves, open into the clearance spaces, which arrangement facilitates the escape of the water of condensation from the cylinder, and obviates the liability to accident. The four valves are moved by one eccentric through the intervention of a wrist-plate; the same valves admit and cut off the steam.

The steam-valve in these engines commences to open its port at one end of the cylinder when the eccentric is producing its most rapid movement, and, as the motion of the eccentric is declining towards the end of the throw, an increasing speed is obtained by means of the wrist-plate, which compensates for the slow motion of the eccentric. At the same time the steam-valve at the opposite end of the cylinder commences to lap its port by the motion of the eccentric, but by a reverse or subtraction of speed produced by the same wrist-plate, which speed is constantly decreasing till the throw of the eccentric is completed. Or in other words, the lapping and opening of the steam-ports require each the same amount of throw of the eccentric, producing, for instance, a lap of ½ an inch at one end of the cylinder, while the opposite end has an opening of one inch and one-eighth. The exhaust-valves are moved by the same eccentric and the same wrist-plate before spoken of, but they have a much greater travel for the purpose of ridding the engine of the exhaust steam easily through the exhaust-ports, which are as long and twice as wide as the steam-ports, and therefore back pressure on the piston of the engine is avoided. The rapid opening and slow lapping of the ex-

haust-ports are obtained in the same manner as in the case of the steam-ports, but much faster, as the travel is greater on the opening of the exhaust than on the opening of the steam-port, in order to get a free and full opening.

The variations of load upon these engines are communicated to the steam-valves instantly by the governor, the valves being moved by a force distinct from it, yet subjected to its regulation. The governor in no case performs any work, and only indicates the changes required to the levers which move the valves, and needs only sufficient force to move a small stop. Its movement is attended with the least possible friction; the stop presents hardly any resistance to the governor, except at the very instant when it is in actual contact with the lever constituting its fulcrum. This momentary resistance by the bearing of the lever on the stop as a fulcrum occupies so small a space of time that, compared with the period during which the governor is left free to move the stop, it is practically nothing. As a precautionary measure, a safety stop is connected with the valve-gear, so that in case the governor should become inoperative, and should fail to act, the steam-valves become unhooked, and cannot open, and, as a universal result, the engine is stopped, although the valve in the steam-pipe may be wide open. The valves are circular, and oscillate on fixed bearings in the front and back bonnets. The valve-stems have flat blades, which extend the whole length of the valves in the steam-chest, and to which levers are keyed for the purpose of giving them motion. The valves are fitted in such a manner as to be capable of adjusting themselves to their seats, as their faces and seats become warm. Any one of them can be adjusted independent of the other, and can be removed from the valve-chests by unscrewing four bolts, and withdrawing a key at the point at which it is attached to the lever. The valve-gear of these engines may be worked by hand, even under extreme steam pressure.

The valve-stems of these engines are packed with an improved metallic packing, which is claimed to possess many advantages in

respect to freedom from friction and wear, over hemp, cotton, or any other fibrous substance, for the stems of oscillating or vibrating engines, as illustrated by the following cut:

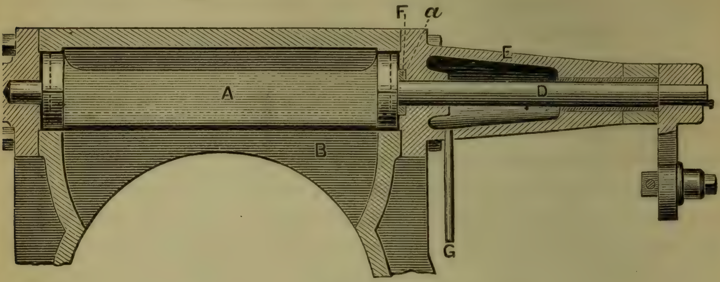


Fig. 1.

A represents the valve, *B* the valve-seat, and *D* the valve-stem or rod, which is outside the chest, and upon the end of which is the toe with which the valve-gear engages to rock the valve to enable the port to be opened. *E* is a standard or bracket projecting from the side of the steam-chest, and bolted thereto, through which the valve-rod passes, and by which the valve-rod, and the valve connected with it, are sustained and supported in their proper relation, all of which is familiar to the construction of steam-engines. At the inner end of the bracket, *E*, and concentric with the hole through which the valve-rod passes, a recess is cut. A collar, *F*, is then shrunk upon the valve-rod, or otherwise tightly fitted thereto, so as to make a flange, and is turned off to face and fit the recess when the valve-rod, valve, and bracket are in their proper relation. The face of the flange, *F*, and the seat of the recess, *a*, should also be round, so as to make a steam-tight joint.

It will be observed from above description that the Harris-Corliss, while retaining all the distinctive features and merits of the original Corliss engine, has in addition the patented improvements of Self-Packing Valve-Stems, Stop Motion on Regulator, Recessed Valve-Seats, Drip Collecting Devices, and Piston Packing.

Questions

FOR ENGINEERS, THE ANSWERS TO WHICH WILL BE FOUND UNDER THE HEADS OF THE RESPECTIVE SUBJECTS TO WHICH THEY RELATE. A PROMPT ANSWER WILL SHOW THAT THE CANDIDATE HAS STUDIED THE SUBJECT, AND IS MASTER OF THE SITUATION, AND VICE VERSÂ.

What are the best aids to candidates applying for an engineer's certificate or license?

What qualifications, both mental and physical, ought candidates applying for a Cadetship in the United States Navy to possess?

What qualifications should candidates for the position of engineers and assistant engineers in the United States Revenue service possess?

What are the necessary qualifications of applicants for the positions of engineers and assistant engineers in the Mercantile Marine service?

What qualifications are necessary for persons taking charge of stationary engines in any locality requiring a license?

Give the names of the different triangles.

What is steam?

What law does the expansive property of steam follow?

What is surcharged steam?

Will it affect the vacuum?

What is saturated steam?

What is supersaturated steam?

To what are the temperature and the elastic force of steam equal?

What is the difference in the pressure of steam when the mercury is in a vacuum, or when exposed to the atmosphere?

If the proper relation of the temperature between the steam and the water from which it is formed be disturbed, what will be the effect?

What is the total heat of steam at 212° Fah.?

How is the latent heat of steam found?

When does heat in steam become latent?

How would you ascertain what amount of water is necessary to condense a given quantity of steam?

What is low-pressure steam?

Why is the steam of salt water fresh?

What is the most extraordinary property of steam?

What are the two modes of applying the power of steam to the cylinders of steam-engines?

State the rule for finding the mean or average pressure in a cylinder.

Is the effluent velocity with which steam of different pressures flows into the atmosphere uniform?

State the difference between the latent and sensible heat of steam at different pressures.

State the total heat and relative volume of steam at different pressures.

As the sensible heat in steam increases, does the latent heat decrease?

In what way does the change affect the economy of the steam-engine?

What is meant by the volume of steam?

What is the difference in volume between water and steam at atmospheric pressure?

How much does one cubic foot of steam at atmospheric pressure weigh?

State the velocity with which steam at different pressures flows into the atmosphere or into steam of a lower pressure.

If steam, at a given pressure, be cut off in the cylinder at a certain point of the stroke, what will be the pressure for the whole length of the stroke?

Give the rule for finding the amount of benefit to be derived from working steam expansively.

Give the rule for finding the average pressure of the steam in the cylinder for different points of cut-off.

Is surcharged steam indicated by the steam-gauge? Or will it affect the vacuum?

What is superheated steam?

What is the steam-jacket?

What is the difference in effect between superheated and saturated steam?

Which is capable of producing the most economical results?

PART SECOND.

Steam-Engines in General.

Steam-engines embrace a great variety of designs and names; such as the beam, side-lever, inclined, oscillating, trunk, horizontal, vertical, and steeple, which are in turn termed single-acting, double-acting, reciprocating, rotary, semi-rotary, compound duplex, inverted, and geared, each of which was probably designed to meet some peculiar requirement — either economy of space, fuel, or efficiency in speed. (Judging from the appearance of things at present, the horizontal and vertical are destined to supersede all other designs for land and marine purposes.)

All steam-engines, of whatever design, or for whatever purpose employed, are embraced under two heads, commonly called high- and low-pressure, but more properly termed condensing and non-condensing. In the non-condensing engine, the steam, after acting on the piston, escapes into the open air; therefore the pressure of the outgoing steam must exceed atmospheric pressure, or 14·7 lbs. to the square inch. Thus, if steam at 45 lbs. average pressure above vacuum be admitted to the piston of a high-pressure engine, it will exert a force equal to its pressure; but 14·7 lbs. per square inch of that pressure will not be converted into work, as it will be lost in overcoming the pressure of the atmosphere, which may be illustrated by the following example:

Diameter of cylinder, 12 in.; area, 113·09 in.
Average steam pressure per square inch, 45 lbs.
Total steam pressure, 5089·05 lbs.

As before, area, 113·09 sq. in.
Atmospheric pressure, 14·7 lbs.
Total atmospheric pressure, 1662·423 lbs.

	5089·050
Loss due to atmospheric pressure,	1662·423
Effective steam pressure on piston,	3426·627 lbs.

The foregoing example shows the resistance to be overcome at each stroke of the piston before the steam acting against it can produce any useful effect. Thus it will be seen that the piston of a high-pressure steam-engine is exposed to the action of two pressures, namely, the pressure of the steam from the boiler on one side, and that due to the atmosphere and the steam remaining in the cylinder after exhaust takes place on the other. The pressure utilized or converted into work will be the difference between the two.

In the low-pressure or condensing engine, the steam, after acting against the piston, escapes into a condenser, where it is condensed into water and a vacuum is formed; thus rendering not only a considerable portion of the steam pressure in the boiler, but also the 14·7 lbs. per square inch required in the non-condensing engine to overcome the pressure of the air, available as an effective force against the piston, which may be explained as follows:

Diameter of cylinder, 12 in.; area, 113·09 sq. in.	
Average steam pressure per square inch, 45 lbs.	
Total effective steam pressure,	5089·05 lbs.
As before, area, 113·09 sq. in.	
Vacuum at best, 13 lbs.	
Power due to vacuum,	1470·17 lbs.
	3958·15
	1470·17
Total effective pressure due to steam and vacuum,	5428·32 lbs.

The back pressure in the condenser, which represents the difference between the indications of the vacuum-gauge and a perfect vacuum, must be deducted; but, as a perfect vacuum is not attainable, the back pressure varies from 2 to 5 lbs., according to the condition of the engine and the quantity of uncondensed steam remaining in the condenser.

Waste in the high-pressure engine.—In the best types of modern high-pressure engines, the useful effect obtained from the work stored up in good fuel may be calculated as follows:

Loss through bad firing and incomplete combustion,	10	per cent.
Carried off by draught through chimney,	30	" "
Carried away in the exhaust steam,	50	" "
Utilized in motive power (indicated),	10	" "
	100	per cent.

The foregoing may seem incredible, and yet any one wishing to do so may demonstrate its truthfulness to his own satisfaction by placing a thermometer in the steam-pipe and noting its temperature during its escape from the boiler to the cylinder; then placing it in the exhaust-pipe, close to the engine, and noting the temperature at this point, when it will be discovered that the steam has lost very little of its heat in passing through the cylinder. Consequently the difference in temperature between the steam when it escapes from the boiler and from the exhaust-pipe, constitutes all of the heat that was contained in the fuel that was utilized.

Waste in the low-pressure or condensing engine.—According to the dynamic theory of heat, as shown on page 105, a certain weight of coal contains within itself a certain amount of work stored up, and ready to rush out under the necessary surroundings, as in the case of a compressed spring set free. The supply of a given weight of coal to the furnace of a steam-boiler represents the application of a definite amount of force at one end of a series of transformations, a part of which force at length appears as useful work at the other, the balance having been wasted in the various processes through which it has passed.

Take, for example, a modern marine-engine of the best construction and design. The force supplied to the furnace in the combustible is first developed as heat by the burning of the coal; a portion of this heat is utilized in changing the water into steam, the balance being wasted either in radiation, or by being carried off in the hot gases through the chimney. A part of the steam formed is applied to move the piston, the remainder being wasted by condensation against the sides of the pipes and cylin-

ders, and by leakage past the piston or valves into the condenser.

It is thus shown that only a small portion of the total force contained in the steam that is applied to move the piston is utilized. Of the force that is utilized in the cylinder, only a small portion performs any external work, the remainder being absorbed in overcoming the back pressure induced by the friction of the machine itself. Of the remaining small portion that may be applied to the screw, another part is wasted in overcoming its useless resistances, and only the balance used to propel the ship.

		PER CENT.
Total heat in one hundred lbs. of anthracite coal, in units of heat............	1,400,000	
Deduct heat equivalent to weight of ashes..................................	200,000	
Total heat in one hundred lbs. of anthracite coal..............................	1,200,000	100
Carried off by hot gases in chimney...	200,000	$16\frac{2}{3}$
Available to produce steam.............	1,000,000	$83\frac{1}{3}$
Lost by leakage and condensation.....	200,000	$16\frac{2}{3}$
Available to perform work in cylinder.	800,000	$66\frac{2}{3}$
Escaped with steam into condenser.....	660,000	55
Transformed into work....................	140,000	$11\frac{2}{3}$
Absorbed in overcoming resistance of engine and load..........................	40,000	$3\frac{1}{3}$
Available to turn the propeller.........	100,000	$8\frac{1}{3}$
Absorbed in useless resistance of the screw.................	20,000	$1\frac{2}{3}$
Usefully applied to propel the ship.....	80,000	$6\frac{2}{3}$

The following figures represent approximately the supposed

distribution of the total force in the best engines. No estimate is taken in them of the coal that is consumed in various ways on board ship other than those mentioned, as, for instance, in getting up steam, or in steam used to work pumps, in steam lost through the safety-valves, in heat lost in blowing off, or remaining in the furnaces after the vessel has arrived in port. These and other causes usually add at least ten per cent. to the consumption, leaving the force utilized about six per cent. of the total force expended in the coal. The cost of an indicated horse-power, by the figures, would be $2\frac{1}{2}$ lbs. of coal per hour nearly.

To mitigate as far as possible the foregoing losses, the surface-condenser and boiler arrangements should be designed so as to insure a rapid circulation of the water, that condition being of the greatest importance to produce efficiency in the heating or cooling surfaces. Whenever practicable, the feed-water should be used as injection to condense the educted steam, so that it might be heated to the highest possible point before being sent into the boiler.

It is a question of vital importance to the owner of a steamship, that the consumption of fuel be reduced to the lowest possible amount, as each ton of fuel excludes a ton of cargo. As improvements in the form of the hull and machinery are effected, less power and less fuel will be required to propel a vessel through the water a given distance; but, great as have been the improvements effected in marine engines to this end, much yet remains to be accomplished, as while the consumption of fuel has been reduced, by working steam more expansively in vessels of later date, from three or four to less than two pounds per effective horse-power, yet, comparing this with the total amount of energy in two pounds of coal, it will be found that not a tenth part of the power is obtained which that amount of coal would theoretically call into action.

To find the quantity of coal required to drive a steamship a given number of days in average fair weather. The beam in feet squared will give the quantity of coal required in net tons. But

it must be understood that, with equal beams, the displacement, and consequently the power required, vary very much; or, in other words, the displacement is not always proportionate to the beam in vessels of the same model, nor is the power required to propel them always proportionate to the displacement.

When steamships are running through still water and still air, the loss due to the resistance of the atmosphere is about ten per cent. of the whole power expended, ninety per cent. being absorbed in overcoming the resistance of the water.

Economy of the condensing over the non-condensing engine.—When the resistance of the atmosphere is removed from the piston, the steam may be cut off earlier, and further expanded in the cylinder. This reduces the draught on the boiler, and admits of a slower combustion of the fuel. In this way economy is promoted by condensation of the exhaust steam and by the vacuum formed in the cylinder. A vacuum equal to 14 lbs. per square inch is 35 per cent. saving in fuel, or the same increase in power; but this saving undergoes a great reduction, in consequence of the cylinder being open alternately to the lower temperature in the condenser, which varies with the degree of expansion employed, being least when the steam follows full stroke, which is very seldom the case. The practical gain, therefore, in the condensing engine is from 20 to 30 per cent., varying with the conditions above named, as shown in the working of condensing engines, both stationary and marine. The economy of the condensing engine might be increased, if advantage could be taken (as in the case of the injector and steam-jet) of the velocity with which the exhaust steam escapes from the cylinder to the condenser. On entering the condenser, the power due to its energy is entirely destroyed by the cold water injection, or by being brought in contact with refrigerating surfaces.

Economy in modern steam-engines, condensing and non-condensing.—In Watt's time, 1 cubic foot or 62½ lbs. of water was the allowance per horse-power per hour for average engines, but the water consumption for most engines was from 75 to 80 lbs.;

while the better class of modern automatic cut-off high-pressure engines will yield a horse-power from a water consumption of from 20 to 25 lbs., and in the best class of condensing engines of from 18 to 22 lbs.; but, in either case, the water consumption depends a good deal on the size of the engines, and the excellence of the design and workmanship, quality of steam, pressure, etc. The last condition exerts a very important influence on the quantity of water required to develop a horse-power.

The mean effective pressure on the piston of a steam-engine is the exponent of the work performed. The term "effective pressure" means the amount by which the total pressure behind the piston exceeds that which acts on the other side in opposition to its movement. The "*terminal* pressure," or that at which the steam releases itself from the cylinder, is the corresponding exponent of the consumption of water by the engine, or the cost of the power. Hence, the best economy is attained when the mean effective pressure is highest relatively to the terminal pressure; and anything that will increase the former without correspondingly increasing the latter, or which will diminish the latter without correspondingly diminishing the former, will improve the economy.

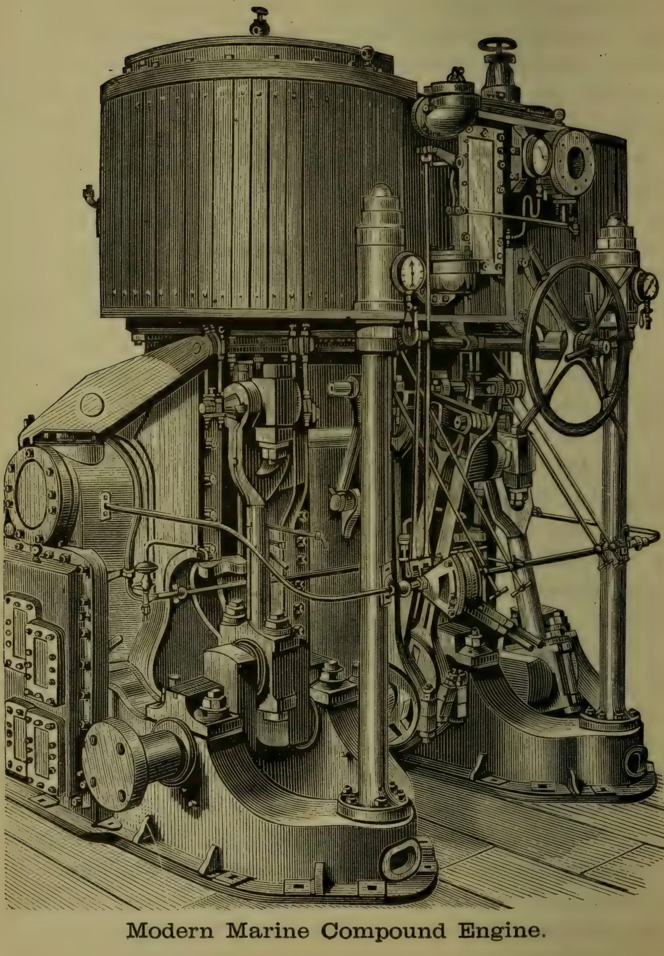
The difference in effect between the condensing and non-condensing engine, with equal pressure of steam and expansion, is solely that the condensing engine has the advantage of the effect produced by the vacuum, or the amount of atmospheric pressure removed. Their difference in operation is, that in the condensing engine, the steam, after having performed its duty in the cylinder, is condensed by the admission of a spray of cold water, or being brought in contact with cooling surfaces, thus producing a vacuum or *minus* pressure, which varies, according to the perfection of the machinery, from 10 to 13 lbs. per square in.; while in the non-condensing engine, the steam, after having performed its duty, is discharged into the atmosphere. Thus, the advantages of the vacuum are lost; some of the waste heat, however, is utilized by leading the exhaust steam through a heater, for the purpose of heating the feed-water.

Compound Engines.

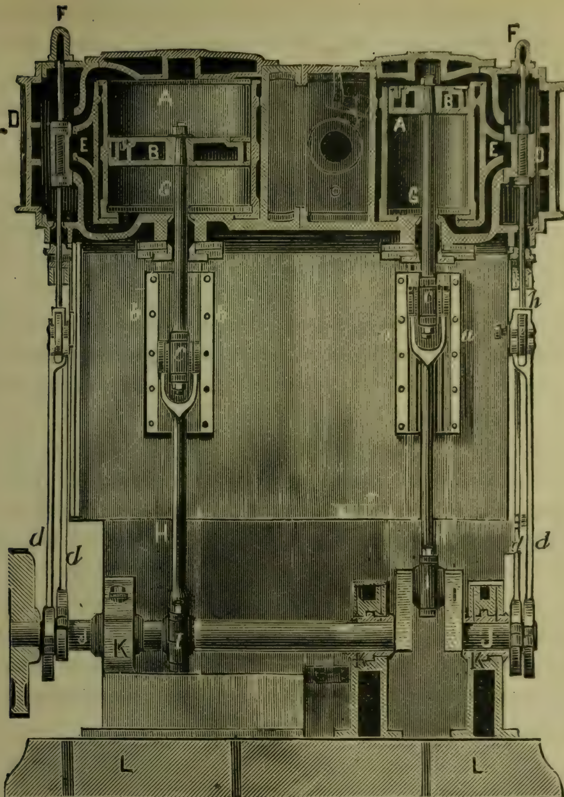
A compound engine is a high- and low-pressure condensing engine, with two cylinders and pistons. The steam is first admitted to the small or high-pressure cylinder, until the piston has moved through a certain distance, when the valve is so regulated, that the communication with the boiler is cut off, the remainder of the space to be passed through by the piston being performed by the expansion of the steam, which, having done its work, escapes to the condensing cylinder, where it does a proportionate amount of work, and out of which it escapes into the condenser.

With respect to the number and arrangements of the cranks and cylinders of compound engines, there are five or six designs used for screw propulsion; but the most generally adopted is the inverted, vertical, direct-acting, with the cylinders both high- and low-pressure, placed alongside of each other in the fore and aft direction of the ship, and the steam-chests between them connected direct with the two cranks on the shaft beneath. Another kind is that of an inverted, direct-acting engine, with one cylinder placed above the other, the high-pressure being uppermost. In this case there is only one piston-rod, which is continued through both cylinders and pistons, one connecting and consequently only one crank, a fly-wheel being generally employed to assist the engine in passing over the centres. In another type, which is known as the HUNTOON, the high-pressure cylinder is placed within the low-pressure. In another, known as the SMART engine, there are four cylinders — two high-pressure and two low; the two high-pressure cylinders being placed on the tops of the low ones, the piston-rods passing through both cylinders and connected directly with the cranks. Lastly, the design most generally adopted for war-vessels is the horizontal, in which the cylinders are placed side by side.

It is claimed that in the better class of compound engines 2 lbs. of coal will develop a horse-power; but the want of reliable data to the contrary would warrant the assertion that 3 lbs. of



Modern Marine Compound Engine.



The annexed cut represents the section through the cylinders, steam-chests, cross-heads, pillow-blocks, etc., of a Compound Marine Engine. *A*, *A*, show the high- and low-pressure cylinders; *B*, *B*, the pistons; *G*, *G*, the piston-rods; *D*, *D*, the steam-chests; *E*, *E*, the exhaust cavities; *F*, *F*, valve-rod guides; *H*, *H*, connecting-rods; *I*, *I*, cranks; *J*, *J*, crank-shaft; *K*, *K*, *K*, pillow-blocks; *L*, *L*, foundation-plate; *c*, *c*, cross-heads; *a*, *a*, *b*, *b*, cross-head-guides; *d*, *d*, *d*, *d*, eccentrics. The steam is admitted from the boilers to the steam-chest of the high-pressure cylinder, from which it is exhausted into the receiver and readmitted into the low-pressure cylinder, after which it escapes to the condenser.

coal are oftener consumed in the development of a horse-power than 2 lbs. Taking 3 lbs. as equivalent to a horse-power per hour, theoretically only about one-sixteenth part has been utilized. The advantages of compound over simple engines is an open and unsettled question, as it is claimed that some simple engines in use at the present time are more economical in the use of fuel than compound engines; but economy of fuel is not the only consideration which leads to a choice of the compound engine for marine service, since, perhaps, the more equal distribution of the power throughout the stroke is a feature of value in these as in all other engines where the resistance is devoid of the controlling influence of the fly-wheel. The disadvantages of compound engines are their extreme first cost, extra weight, and complication of parts.

The receiver is a chamber between the cylinders of compound engines into which the steam from the high-pressure cylinder escapes, and from which it is admitted to the low-pressure cylinder. The receiver may be said to be the steam-drum for a low-pressure cylinder; its capacity might be infinite, except for the weight and expense it would incur. In the majority of independent compound engines, the capacity of the receiver is about equal to that of the low-pressure cylinder, though for engines in general its capacity is regulated by certain attending circumstances. Such engines as the Worthington have no receivers.

Simple Engines.

All steam-engines are divisible into two classes, simple and compound; the latter being those in which the steam is used twice, by being exhausted from one cylinder into another, while the former applies to all engines which use steam only once, whether they are double engines and have double sets of valve-gear or not. The term single engine is sometimes used; but it is liable to give rise to confusion.

Locomotives, steam fire-engines, and stationary engines which

take their steam directly from the boiler and exhaust it into the atmosphere should be termed simple engines, regardless of the number of cylinders. An impression very generally prevails among engineers that compound engines are of necessity marine engines, and also condensing, which is a mistake, as there are both high-pressure and low-pressure, or condensing and non-condensing compound engines. Condensing compound engines generally have not more than two cylinders, although in some instances they have three, while non-condensing compound engines are met with which have four.

Marine Engine. — The term marine engine is in very common use, but it has no definite meaning, as it may be either condensing or non-condensing, vertical, horizontal, or inclined, simple or compound. The only reason that can be assigned for designating it a marine engine is that it was designed to be used on steam-ships. A marine engine, properly speaking, is an engine designed to occupy a certain space on a vessel, and be capable of developing a certain amount of power. The most desirable class of marine engines are those that develop the greatest amount of power with a given area of piston and steam pressure, and that occupy the least space. The vertical engine is more in favor with marine engineers, as it possesses many advantages over any other design. This perhaps arises from the fact that it occupies less floor space; that it is more compact, and less liable to spring than an engine of any other design; and that the weight is against the lifting-force of the reciprocating and revolving mechanism; also that, in consequence of the housing and pillow-block bearings being in one piece, they are less liable to get out of line than those of any other arrangement; and that they afford better facilities for a direct connection with the propeller shaft than any other. Trunk and oscillating engines are still employed in England for marine purposes, but only on war-vessels. Such designs never were looked upon with much favor by enlightened engineers.

TABLE

OF THE AVERAGE PERFORMANCES OF DIFFERENT DESIGNS OF PUMPING-ENGINES.

Location.	Date.	Engine.	Designer.	Duty per 100 Lbs. of Coal.
United Mines, Cornwall..	Sept., 1842.	Cornish single cylinder, jacketed......	Taylor................	114,361,700
Carn Brea, Cornwall......	1841.	Cornish compound, jacketed...........	James Sims.........	101,702,000
Lynn, Mass.................	Dec. 1873.	Compound beam and fly-wheel, jacketed...........................	E. D. Leavitt	103,923,215
Lowell, Mass................	June 1875.	Compound beam and fly-wheel, jacketed...........................	Simpson............	117,350,100
Lawrence, Mass............	May 1876.	Compound beam and fly-wheel, jacketed...........................	E. D. Leavitt	96,201,900
Trenton, N. J	Mar. 1876.	Compound beam and fly-wheel, jacketed...........................	Wm. Wright.......	84,500,000
Milwaukee, Wis...........	May 1875.	Compound beam and fly-wheel, jacketed...........................	R. W. Hamilton...	76,955,720
Marion, Ind.................	Feb. 1877.	Single cylinder yoke and fly-wheel condensing..................	Dean.................	49,231,207
Haarlem Meer, Holland.	June 1848.	Compound beam annular cylinder.....	Gibbs & Dean......	80,000,000
Chicago......................	Dec. 1874.	Single cylinder beam and fly-wheel, unjacketed.................	D. C. Cregier	65,824,581
Chicago......................	April 1877.	Compound beam and fly-wheel, unjacketed.........................	Quintard Works...	West engine 99,083,300
Chicago......................	April 1877.	Compound beam and fly-wheel, unjacketed.........................	Quintard Works...	East engine 96,066,800
Chicago......................	April 1877.	Compound beam and fly-wheel, unjacketed.........................	Quintard Works...	75,000,000

TABLE

OF THE AVERAGE PERFORMANCES OF DIFFERENT DESIGNS OF PUMPING-ENGINES.

LOCATION.	DATE.	ENGINE.	DESIGNER.	DUTY PER 100 LBS. OF COAL.
Cincinnati	Nov. 1872.	Horizontal crank and fly-wheel, two engines coupled non-cond.	Shields	43,566,178
Cincinnati	Nov. 1872.	Vertical single cylinder crank and fly-wheel condensing	Scowden	37,789,990
Cincinnati	Nov. 1872.	Vertical single cylinder crank and fly-wheel condensing	Scowden	34,064,977
Cincinnati	Nov. 1872.	Vertical direct acting single cylinder condensing	Shields	23,580,687
Louisville	1873.	Cornish	Scowden	37,536,730
Newark, N. J	1870.	Compound duplex	Worthington	77,157,840
Cleveland, Ohio	1873.	Cornish	Allaire Works	41,774,955
Jersey City	1856.	Cornish	W. Point Foundry.	72,115,396
Charleston, Mass	1872.	Duplex	Worthington	56,937,643
Providence	1874.	Radial cut-off	Geo. H. Corliss	25,865,000
Providence	1874.	Compound duplex	Worthington	53,528,210
New Bedford, Mass	1869.	Beam and fly-wheel	McAlpine	59,336,497
Brooklyn, No. 1	1860.	Single cylinder beam	Wright	60,798,200
Cleveland, Ohio	1875.	Compound duplex	Henderson	31,968,006
Toledo, Ohio	Sept. 1875.	Compound duplex	Worthington	45,611,924
Columbus, Ohio	Feb. 1876.	Crank and fly-wheel, four engines coupled	B. Holly	24,045,951

Uncertainty of Tests made for the Purpose of Comparing the Relative Economy of Marine Engines.

It has been customary heretofore, in order to determine the relative economy of marine engines, to weigh the amount of coal consumed in performing a certain amount of work. So long as all the machines compared are of the same design and dimensions, the coal used of the same kind and quality, and the pressure of the steam, the degree of vacuum, the rate of expansion, the temperature of the atmosphere, and all other circumstances are the same, it may be inferred that any difference in the economy is the result of some imperfection in the engine itself. But if there is a variation in one particular only, as, for instance, in the degree of vacuum, the difference may be assumed to be due to that variation; but if there are several variations at the same time, where there are different kinds of engines or boilers and different steam-pressures, when there is any gain or loss of economy, it is impossible to decide to which of the variations the change is due. So, also, where high pressure of steam is carried, and a greater expansion is employed, if a poor economy is shown, it may happen that the benefits that should result from the high pressure and the increased expansion were counteracted by the increased condensation and leakage, or that the power which was gained in the engine was lost in the boiler, or *vice versâ.* Then, again, any difference in the kind of fuel employed, or in the skill and management, unite with the other variations to render the actual results more unsatisfactory.

In attempting to compare the results of such experiments as are recorded to determine the most economical design of engine, it will be generally found that the experiments made to determine one certain point are not sufficiently complete to serve any other purpose, and have generally been made by different men, under different circumstances and in different localities; and, moreover, that the expert is almost invariably biassed by prejudice. This is particularly applicable to the reports which may be obtained

of the performances of new and improved engines, which may be accounted for in this way: A steamship company may place on its lines a vessel of fine model, with the most improved machinery, which, on comparison, would show more satisfactory results than one of the same capacity but of inferior lines, and propelled by an inferior style of engine. It will be found, on comparison, that the profits resulting from the new ship and improved engines are largely in excess of those of the old; but it would, at the same time, be difficult to separate the gain due to the improvement in the model of the hull from that due to the improved engines, and *vice versâ.* All that can be done in such cases is to accept the whole result, without being able to separate the one from the other. A series of exhaustive experiments to determine the relative economy of different classes of steam-engines and boilers is very much needed, but the difficulties to be encountered are so numerous as to render such an undertaking impracticable.

Power of steam-engines.—The power which a steam-engine can furnish is generally expressed in "horse-power," the "nominal horse-power" being admitted to be a force capable of raising a weight of 33,000 pounds * one foot high in one minute, or 150 pounds 220 feet high in the same length of time. If an engine is rated at 25 horse-power, it is recognized as being capable of raising 33,000 pounds one foot high twenty-five times in each minute. The question will naturally arise, How are these 33,000 pounds to be raised? The answer to which would be, by *belts, pulleys, cog-gearing, cables, paddle-wheels, screw-propellers,* or *whatever mechanical arrangement* is most practicable and convenient.

There are several terms employed to express the power of engines, such as the "nominal," "indicated," "actual or net," "dynamo-metrical," and "commercial" horse-power. The *indicated horse-power* is obtained by multiplying together the mean effective pressure in the cylinder as shown by the diagram, the area of the piston in square inches, and the speed in feet per minute, and

* Foot-pounds.

dividing the product by 33,000. The *actual* or *net horse-power* is the total available power of an engine; it equals the indicated horse-power less the amount expended in overcoming the friction. The *dynamo-metrical horse-power* is the net horse-power after allowing for friction. The term *commercial horse-power* is sometimes used, but has no definite meaning, as there is no recognized rule among engineers by which to buy or sell engines.

Estimating the power of steam-engines.—There are three conditions necessary to be understood, before we can calculate with any degree of accuracy the power which a steam-engine is capable of developing—*first*, the number of square inches in the piston; *second*, the effective pressure exerted against each square inch of the same; *third*, the speed of the piston in feet per minute. Nor can the power which a steam-engine is exerting be demonstrated by any calculation, however accurate, unless the condition of the engine and the back pressure be also known, which latter can only be determined by the *indicator*.

How to increase the power of steam-engines.—The three most practical methods of increasing the power of a steam-engine are, 1st. To enlarge the diameter of the cylinder; 2d. To increase the speed; 3d. To increase the pressure of the steam. But the increase in any case must have a very narrow limit, as, if the diameter of the cylinder be increased much, the other parts of the engine will be too light. The steam pressure cannot be increased more than the boiler can safely bear, nor can the speed be increased beyond what the revolving and reciprocating parts of the engine will bear. But the power of any high-pressure engine can be very materially increased by attaching a condenser and an air-pump to it, providing the water supply is sufficient.

Speed of engines.—The speed of steam-engines is generally counted by strokes, one stroke being half a revolution, or one revolution being two strokes. The crank travels from one dead-centre to the other to make one stroke, the distance travelled by the crank-pin while making a stroke being twice the distance between the centres of the crank-pin and crank-shaft. To find the

travel of piston in feet per minute, multiply the distance travelled for one stroke in inches by the whole number of strokes in inches, and divide by 12.

Over-stroke.—This term is used when the position of the piston in the cylinder is so altered by taking up the lost motion in the boxes that it strikes either cylinder-head when the crank is at the dead-centre.

The Locomotive.*

In estimating the power of a locomotive, the term horse-power is not generally used, as the difference between a stationary steam-engine and a locomotive is, that while the stationary engine raises its load, or overcomes any directly opposing resistance with an effect due to its capacity of cylinder, the load of a locomotive is drawn, and its resistance must be adapted to the simple adhesion of the engine, which is the measure of friction between the tires of the driving-wheels and the surface of the rails.

The power of the locomotive is estimated in the moving force at the tread of the tires. It is called the tractive force, and is equivalent to the load the locomotive could raise out of a pit by means of a rope passed over a pulley and attached to the circumference of the tire of one of the driving-wheels. The adhesive power of a locomotive is the power of the engine derived from the weight on its driving-wheels, and their friction or adhesion to the rails.

If the wheels of a locomotive were geared into toothed rails, its power would be the force with which its wheels could be made to turn, or the weight or force which, if applied at the rims of the wheels, would prevent them from turning. But if the wheels revolve on smooth rails and slip in turning, a part of the power would be wasted, and the effective power of the engine limited by the friction or adhesion of its driving-wheels. Hence the terms

* For full particulars on this subject, see Roper's "Hand-Book of the Locomotive."

"tractive power" and "adhesive power" mean respectively the revolving power and the progressive power of the engine.

The Steam Fire-Engine.*

Steam fire-engines are simply hydraulic machines similar to steam-pumps, and the conditions involved in their employment are precisely the same. They are also steam-engines, with their machinery adapted to a special purpose, it being perfectly immaterial whether they are movable or stationary. Their means of locomotion is only a matter of convenience. The result of the working of the steam fire-engine may be measured by the hydraulic effect, and the power utilized may be determined by the quantity of water delivered.

To determine the efficiency of steam fire-engines, it is necessary to note—*first,* the extreme vertical height and horizontal distance to which the water can be thrown; *second,* the volume or quantity delivered in a certain time; *third,* the total power consumed in performing that work.

Rule *for finding the horse-power of steam-engines.*

Multiply the area of the piston in inches by the average steam pressure in pounds per square inch; multiply this product by the travel of the piston in feet per minute,† and divide this product by 33,000; the quotient is the horse-power.

Rule *for finding the horse-power of steam fire-engines.*

Multiply the area of the piston in inches by the steam pressure in pounds per square inch; multiply this product by the travel of the piston in feet per minute, and divide this last product by 33,000; ·7 of the quotient will be the horse-power.

Rule *for finding the horse-power of a locomotive.*

Multiply the area of the piston in inches by the pressure in

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* For a full description of all the steam fire-engines in use at the present day, their peculiarities of design, construction, efficiency, etc., see Roper's "Hand-Book of Modern Steam Fire-Engines."

† Which should never be less than 250 feet per minute; in fact, that should be the minimum piston speed for all classes of engines.

pounds per square inch; multiply this product by the number of revolutions per minute; multiply this by twice the length of the stroke in feet or inches; multiply this last product by 2 and divide by 33,000; the result will be the horse-power.

Rule *for finding the horse-power of simple condensing engines.*

Multiply the area of the piston in inches by the mean effective pressure in pounds per square inch; multiply this product by the velocity of the piston in feet per minute; multiply the atmospheric pressure in pounds per square inch on the bucket of the air-pump by its velocity in feet per minute; subtract the last product from the second, and divide the remainder by 33,000; the quotient will be the horse-power of the engine.

In estimating the horse-power of steam-engines by the foregoing rules, not more than two-thirds of the boiler pressure should be taken; as the analysis of a large number of indicator diagrams shows that the average pressure in the cylinders of slide-valve engines rarely, if ever, exceeds two-thirds of the boiler pressure. This difference is due to the reduction caused by the pipes, stop-valves, and the condensation in the pipes, cylinder, etc.

Rule *for finding the horse-power of a steam-engine by indicator diagrams.*

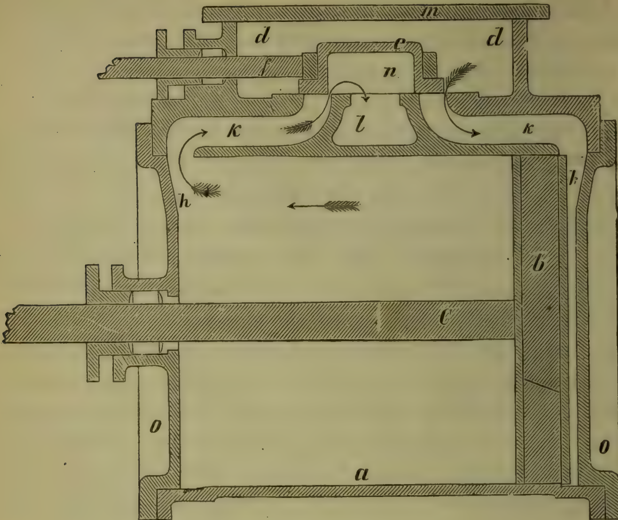
Multiply the area of the piston by its travel in feet per minute, and divide by 33,000; the quotient will be the value of one pound of mean effective pressure, which, if multiplied by the total mean effective pressure, as shown by the card, will give the indicated horse-power.

Example.—Area of piston, 113.

Travel of piston in feet per minute, $333\frac{1}{3}$.

$$\frac{113 \times 333\frac{1}{3}}{33{,}000} = 1{\cdot}141 \text{ horse-power value of 1 lb. M. E. P.}$$

$$\begin{array}{rl} 36 & \text{M. E. P. as shown by the card.} \\ \hline 6846 & \\ 3423 & \\ \hline 41{\cdot}076 & \text{horse-power.} \end{array}$$



The above cut shows a section of the cylinder, piston, and steam-chest of an ordinary slide-valve engine; *a* represents the cylinder; *b*, the piston; *c*, the piston-rod; *o o*, recesses in the cylinder-head; *k k*, steam-ports; *l*, exhaust cavity in the valve-seat; *n*, exhaust opening in valve-face; *e*, valve; *f*, valve-rod; *d d*, steam-chest; *m*, bonnet of steam-chest, and *h h*, clearance.

The term clearance is understood by engineers to mean the unoccupied space between the piston- and cylinder-heads when the crank is at the dead-centre; but it also applies to the space between the cylinder and the face of the valve or valves, either slide or poppet. The amount of clearance of any engine affects its economy; and if the clearance is small, the engine will be more economical than if large; a certain amount is an absolute necessity. It is, therefore, an object of importance, in point of economy, to have the valve-face as near the base of the cylinder as possible. In this lies one of the most important features of the Buckeye,

Brown, Putnam, Woodruff and Beach, etc., and, in fact, all engines of the Corliss type. The clearance varies with different builders, and in different engines from $1\frac{1}{2}$ to 10 per cent. of the cubic contents of the cylinder.

The clearance is often as high as fifteen per cent., in some old-fashioned long stroke, slide-valve engines. This arose from a misconception, at the time they were designed, of the waste the large clearance would occasion, and is, perhaps, in many instances, due to the caprice of the inventor of some patent piston, who made his piston-rings of less depth than the original designs, thus increasing the space between the piston- and cylinder-heads, when the crank is at the dead-centre. There are even cases to be met with, where the old fashioned, hemp-packed piston has been replaced by metallic packing of not more than half its depth, without any means being taken to fill up the spaces at each end of the cylinder. Now, providing that the clearance is fifteen per cent. of the cubic contents of the cylinder, and that the engine makes from one hundred and fifty to two hundred strokes per minute for ten hours, it may easily be seen how enormous the waste must be. The quantity of fuel that might be saved by replacing such an engine by one in which the clearance would be reduced to a minimum, would more than pay for the latter in five years. Persons employing steam-power, or intending to purchase steam-engines, should pay attention to the foregoing fact.

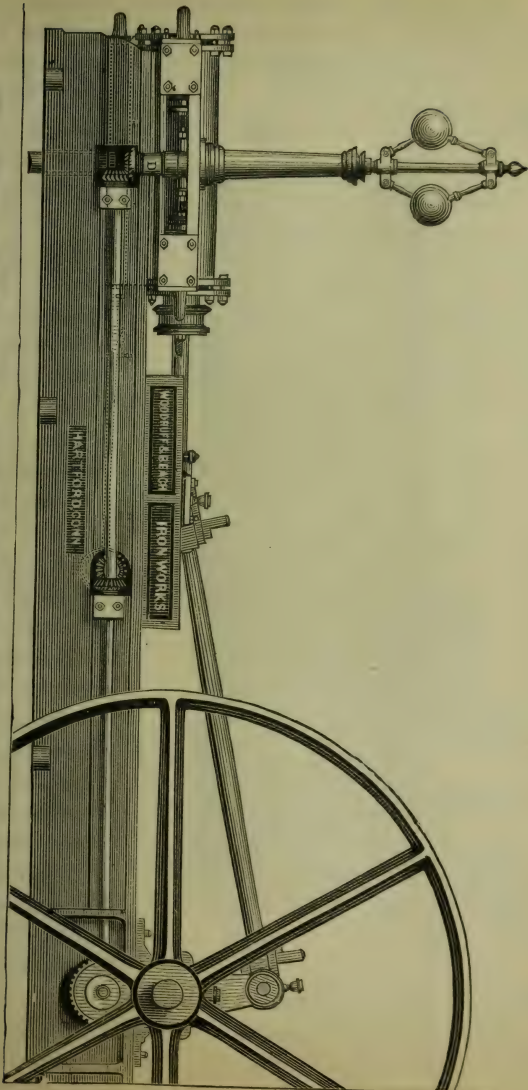
As the clearance space is generally irregular in form, particularly in slide-valve engines, it is somewhat difficult to calculate the exact cubic space. The most accurate method of ascertaining the exact amount of the clearance is to place the crank at the dead-centre, and fill the space with water up to the face of the valve (the quantity of water being previously weighed or measured). Then deduct the amount remaining in the vessel from the whole, and the remainder will be the quantity contained in the clearance in cubic inches or gallons, as the case may be.

The Woodruff & Beach Automatic Cut-Off High-Pressure Engine.

The cut on the opposite page represents the Woodruff & Beach high-pressure automatic cut-off engine. Fig. 1 shows a section of the cylinder-valves, steam passages, and exhaust passages. Fig. 2 is a back view of the cylinder, steam-chests, valve-gear, etc. With the exception of the Corliss, it is the oldest variable cut-off engine in the country, and one that has undergone fewer changes in its mechanism than any other. Those who remember it thirty years ago, will fail, at the present day, to discover much difference from its general appearance. For more than a quarter of a century it has successfully competed with such engines as the Corliss, and it has always sustained a high rating in the scale of comparative merit. The bed-plate, as will be observed, is of the ordinary box O. G. pattern, to which the cylinder-guides and pillow-blocks are bolted and dowelled in such a manner that the possibility of their getting out of line is entirely obviated.

The steam-valves, which are of the double poppet form with bevelled faces and seats, are located at the back of the cylinder at each end, horizontal with its axis. Their stems project inward, and, owing to the peculiar shape of the cam which gives the motion, the opening and closing is done very quickly and almost noiselessly. They have independent adjustments, so that the steam lead may be varied to meet any requirement without interfering with the rest of the valve-gear. The power required to work the valves in these engines is very slight, and as the cam-lug and the ends of the valve-stems are made of hardened steel, they show no perceptible sign of wear after years of use.

The exhaust-valve, which is cylindrical in form and has a very convenient arrangement for taking up the wear and preventing leakage, is placed at the bottom of the cylinder, and communicates with it by its own ports or passages, which are entirely separate from those of the steam-valve. An equilibrium of pressure is maintained by the exhaust taking place through the interior



The Woodruff & Beach Automatic Cut-Off High-Pressure Engine.

of the valve, and as its stroke is very short, the liability to wear is slight. Its motion is derived from a transverse shaft under the centre of the guides, carrying an eccentric, and driven by bevel-gears. Owing to the position of the exhaust openings at the bottom of the cylinder, and their ample size, the exhaust is very free; the discharge of any water that may accumulate from condensation or from priming in the boilers is rendered easy, and all danger of accident from this cause is obviated.

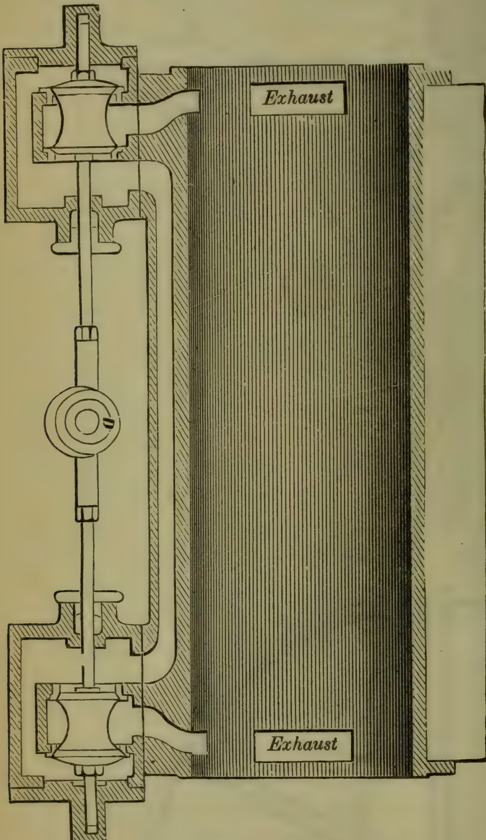


Fig. 1.

The governor, which is very powerful and sensitive, and of a kind that is admirably adapted to these engines, is located centrally between the steam-valves, and receives its motion from a longitudinal shaft, supported on bearings attached to the bed-plate, and driven by a spur and bevel-gear from the crank-shaft. Its spindle passes through a compound eccentric carrying a movable cam-lug, which, by its rotation, gives the opening or

outward motion to the valves, in which direction it is positive; while the closing, although controlled by the cam, is effected by the pressure of the steam upon the unbalanced area exposed at the outer end, and is assisted by a spiral spring. In the bore of the inner eccentric is an inclined or spiral slot for the reception of a key attached to the governor-spindle, from which the eccentric receives its motion. As the key is raised or lowered by the variations of the governor, the inner eccentric is turned to the right or to the left, and the cam-lug moved in or out, as the case may be, thereby giving the necessary opening to the valve, and cutting off the steam at the right point to allow of the proper degree of expansion. As the cam-lug is at all times in the same relative position to the outer shell of the eccentric, the lead of the steam-valve is not affected by the variations.

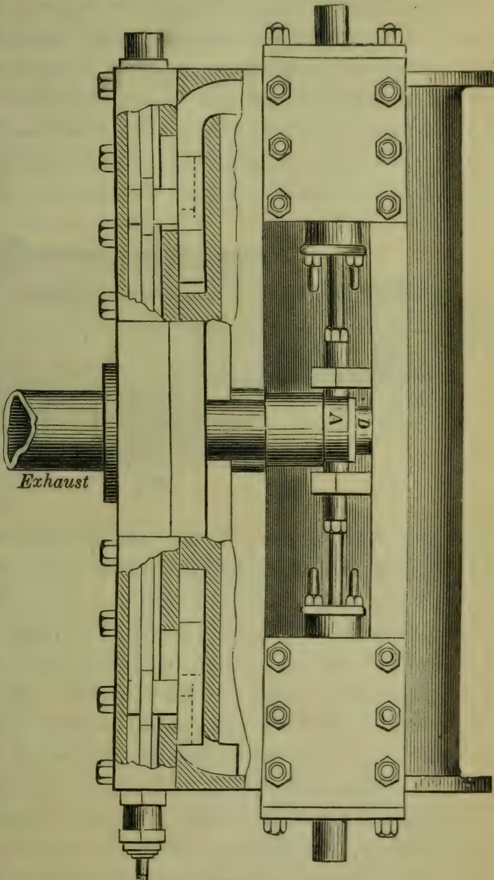


Fig. 2.

The expansion-gear of these engines is one of the most ingenious, simple, and effective mechanical devices that can be employed for that purpose. Its operation may be explained as follows: The cam, marked *C*, Fig. 3, cuts off the steam with certainty at any part of the stroke, the motion being produced automatically by the action of the governor upon it, throwing it more or less out of centre with the spindle of the governor; the rotation of the balls being more or less rapid, the eccentricity of the cam determines the amount of steam admitted to the cylinder. To produce this effect the cam is made of two pieces. *C* is a hollow shell or

Fig. 3.

cylinder, with a part of one end formed into a cam proper. Throughout the whole length of this piece, upon the inside, there is a spiral groove cut to receive one end of a feather, by which its pitch or eccentricity is regulated. The inside piece, *D*, (Fig. 4) is a hub which exactly fits into the hollow of the cylinder, *C*, and has a socket, *e*, into which the spindle of the governor is secured, the other end, *d*, forming a journal or bearing with a bevel-wheel on its extremity, to transmit the motion from the crank-shaft gearing to the governor and cut-off. There is a hole throughout the length of the inside piece, *D*, which is continued through the spindle of the governor, and which contains the rod which connects the cam with the governor. This hole is eccentric to the outside surfaces of *D* and *C*, but is concentric with the collar, *f*, and with the governor-rod. Both pieces, *C* and *D*, are connected by a feather, one piece of which is of a spiral form, and the other straight; the

two being connected together by a stub which fits into a hole or bearing in the spiral piece, so that the latter can turn on the stub and accommodate itself to the groove in which it works. The spiral part of the feather works in a spiral groove in the inside of the shell, *C*, and the rectangular piece works in a straight groove on the inside of the hub, *D*, the inner part of the rectangular piece being fastened to the governor-rod, so that the feather is permanently connected with the governor. When the several pieces are put together the cam is complete, as shown in Fig. 4, and it operates as follows: Motion is communicated by gearing from the crank-shaft to the bevel-wheel on the end of the piece,

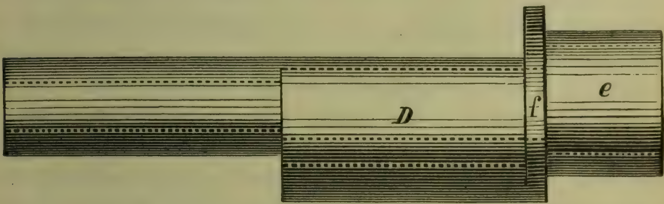


Fig. 4.

D, as well as to the spindle of the governor, which is screwed into the socket on *D*; as the balls rise or fall through change in the centrifugal force, due to the variation in the speed of rotation, they raise or depress the governor-rod, which passes through the spindle, and the piece, *D*, which is attached to the feather thereby raising or depressing it. This feather acting on the spiral groove instantly alters the lift of the cam, and regulates the amount of steam admitted to the cylinder. By these means any speed may be selected at which the load of the engine is to move, and any variation from that will be instantly felt by the governor, and corrected. There is no jar in the working of the parts; the feather moves noiselessly in its grooves; the governor-rod moves up and down through the spindle and the piece *D*, and can be regulated to give any required opening of the steam-ports to suit the work to be done.

The Woodruff & Beach engines are very simple in design, and have the reputation of being very durable and economical. Absence of complication in the valve-motion, and the ease with which all the working parts can be adjusted, are valuable features.

Automatic Cut-Off and Throttling Engines.

All steam-engines, for whatever purpose designed or employed, are either automatic cut-off or throttling. In the automatic cut-off engines, the steam-valves are so controlled by the governor, as to cut off the steam at any point from zero to three-quarter stroke — the cut-off taking place earlier, or later, to accommodate the varying load on the engine and the pressure in the boiler, the object being to obtain full boiler pressure at the commencement of each stroke, and maintain it to the point of cut-off, leaving the balance of the stroke to be completed by expansion, the speed of the engine being controlled by the cut-off and not by throttling. In engines of this class, there is no impediment (save such as may occur at the port of entrance) to the free flow of steam from the boiler to the cylinder, the regulation being effected not by diminishing the pressure, but by cutting off in the cylinder the volume of steam necessary for each particular stroke; consequently, the only loss in pressure between the boiler and the cylinder is that due to the number of bends, and the length of the connecting pipe.

Although all intelligent engineers are agreed upon the superior economy of the automatic cut-off engine, few, excepting those who have had the opportunity of making a practical comparison, are aware of the great saving in the expense of fuel over that class of engines wherein the point of cut-off is invariably relative to the stroke of the piston. It is well understood, that the amount of work realized, as compared to the total theoretical work due the volume of steam expended, even in the most perfect engine, is a very small percentage of the whole energy; and it is, therefore, the more an object of interest to know precisely what the differ-

ence is between these two classes of engines in point of economy. The conditions which insure the highest grades of economy are a full port with no intervening obstructions to impede the free flow of the steam, and a rapid movement of the cut-off, or steam-valve, over the port; as mere increase in the mean effective pressure, resulting from a tardy closing of the port, represents no gain during one stroke of the piston that may be stored up and expended during the succeeding stroke; hence, any force upon the piston in excess of that required to balance the resistance will result in a diminished economy.

The economy of high-pressure engines is exactly in proportion as their average piston pressure is higher than the terminal, providing the latter does not fall below that of the atmosphere; the highest economy being attained when the stroke is commenced with full boiler-pressure, and the steam quickly and completely cut off, at a point in the stroke that allows the pressure to fall to, or very near, that of the atmosphere.

Throttling Engines.

Throttling engines are those in which the flow of steam from the boiler to the cylinder is regulated either by a throttle-valve, a kind of damper in the steam-pipe, which, as the speed of the engine increases, is turned, and stops off the supply of steam, or by the steam in its passage from the boiler to the cylinder oozing through the passage of some peculiar type of governor-valve. An engine controlled by any such device is in a condition somewhat like that of a horse restrained by a brake applied to the wheels of a wagon. Such relics of barbarism are fast giving place to the automatic cut-off arrangement, by which the brakes are removed from the wheels, and the bit placed in the horse's mouth, instead. Manufacturers of this class of engines claim that they give results equal to the automatic cut-off engines, which is untrue, both as to economy and close governing. With an early cut-off, which is absolutely necessary to good economy, it is simply

impossible to govern the speed of throttling engines closely, with even a moderate change in load and pressure.

In the best types of throttling engines, in consequence of the peculiar construction of the governor-valve, and the tortuous passage through which the steam has to travel, the pressure in the cylinder is in many cases not more than one-half of the boiler pressure; the effect of which is, that when the work to be performed is varying in its nature, such engines increase their speed when any considerable load is thrown off, and decrease it when additional load is put on. Now, every stroke an engine makes above its regular speed is a waste of steam, and if the engine is large, or runs at a high speed, the volume of steam, and consequently of fuel, wasted will be enormous; likewise every stroke an engine makes below its ordinary speed, when work is thrown on, lessens production. The loss of one revolution in ten diminishes the productive capacity of every machine driven by the engine 10 per cent.; in short, the loss of one revolution in ten diminishes the productive capacity of the whole factory 10 per cent.; while the expense of conducting the whole business, rent, wages, insurance, etc., continues the same as if everything was in uniform motion. A variation of one revolution in ten is quite common in throttling engines: in fact, it is unavoidable.

Steam-Engine Cut-Offs.

The great desideratum in the use of steam is the most perfect application of the expansive principle. As the pressure of steam is always calculated in pounds per square inch above atmospheric pressure, the nearer the indicated line of expansion approaches that of the atmosphere, the greater is the actual power derived from the utilized volume of steam. Were the boiler pressure and the load or resistance on an engine always uniform, it would be an easy matter, by making the cylinder of the necessary dimensions, to set the cut-off at the proper point for allowing of proper expansion. As, however, the pressure and load are constantly

varying, it is necessary to reduce the consumption of steam to a minimum which, by its perfect expansion, will give the required power. To these considerations may be attributed the efforts which have resulted in the adoption of the three devices now in use, viz., the *positive*, *adjustable*, and *variable*, or automatic cut-offs.

In the *positive* cut-off the expansion of steam is effected by what is known as lap on the valve, by which the steam is cut off at the same point in each stroke, independent of load or pressure; although in some instances the expansion of steam in the cylinder is effected by an independent cut-off riding on the back of the main valve, and receiving its motion from an eccentric. Such an arrangement, like the former, is productive of beneficial results; but nevertheless it is very defective, inasmuch as it is stationary, and cannot be varied to meet the requirements of work, pressure, and speed.

In the *adjustable* cut-off the expansion is effected by an independent valve, which can be adjusted by the engineer, outside of the steam-chest, by means of a screw, hand-wheel, or other mechanical arrangement to meet the requirements of work and pressure. The link, in its application to the steam-engine, belongs to this class of cut-offs. Although such arrangements are adjustable, they are not self-adjusting, and when once set will cut-off independent of circumstances.

The *variable* or automatic cut-off performs its functions according to circumstances of load and pressure, both in admitting and cutting off the steam. It gives regularity of motion and secures all the benefits of expansion, as the governor operates the mechanism which determines the exact point in the stroke where the supply of steam from the boiler should be cut off and expansion begin. This insures the most perfect regulation under the most varying circumstances, as the slightest change in the position of the governor will increase or decrease the initial charge of steam admitted, thus balancing any variation in the amount of resistance. It must not be inferred from the foregoing that any mechanical arrangement that may be termed by its inventor an auto-

matic cut-off is capable of producing economical results, as many of them are nothing but rattle-traps, undeserving of the name of automatic cut-offs.

The cut-offs most generally used on steam-boats, tugs, and ferries, are either the Stevens, Sickles, or Winters. They all receive their motion from an eccentric on the main shaft. The Stevens cut-off has two rock-shafts,— one for the steam and one for the exhaust,—which are operated by two separate eccentrics. The Sickles cut-off is operated by an eccentric, the valves being tripped by a wedge, so arranged as to disengage the valve-gear at any point of the stroke. Dash-pots are employed to ease the valves into their seats. The Winters cut-off is operated by a revolving shaft, which receives its motion from an eccentric. One of the advantages of this cut-off is that it can be arranged to cut off at any desired point of the stroke when the engine is in motion, but neither the Stevens nor the Sickles can. ZACHARIAH ALLEN, of Providence, R. I., was undoubtedly the inventor of, and the first to practically apply, the automatic cut-off, which is unquestionably one of the greatest improvements ever made in the steam-engine.

Design of Steam-Engines.

The design or improvement of any class of machinery must be based upon two suppositions, either that existing mechanism is imperfect in its construction, or that it lacks functions which a new design may supply. In most cases it would seem that any machine, or part thereof, is susceptible of improvement; yet it will be generally found no easy matter to hit upon a design, or conceive a plan, to remedy the existing fault. Therefore no person should undertake to design a machine unless he is well acquainted with the principles involved in working it. He should be able to calculate strength, strains, and forces, and apply the calculations so as to apportion the quantity and form of the material in the various parts of the machine, in order to produce the greatest amount of strength with the least expenditure of material. Be-

sides a design may be based on right principles, and yet unforeseen mechanical difficulties may prevent its application; it may introduce complication of parts, incur extra expense, or not be susceptible of convenient or easy adjustment. The fewer the parts, and the more harmonious the action, the more valuable the machine will be, providing it embodies a good principle in its design.

Before any correct formulæ by which to determine the proper proportions for steam-engines can be deduced, there are many things to be considered, such as permanent load, weight of moving material, nature of motion, etc. The load on the piston-rod consists of the piston at one end, and the cross-head at the other; consequently the greater the length between these two points the more the rod is affected. For this reason, it is obvious that, when it becomes necessary to determine the area of the piston-rod, the pressure area of cylinder load and length of travel must be duly considered. The connecting-rod being hung between a sliding and a rotary motion, the load is in some measure due to the length of the rod in proportion to the circle described. In the first case, the sliding-point has a load on it due to the weight of the piston-rod, beyond the stuffing-box, with the additional weight of the cross-head. In the second instance, the rotating surface is affected by the weight of the rod and that of the crank.

To determine the diameter of the crank-shaft we must take into account the weight of the crank as a lever, and the pressure of steam as the weight on the end of the same. The proportions of the crank-pin are likewise modified according to pressure, permanent load, length of stroke, shearing strain, etc.

The most valuable features of a steam-engine are strength, durability, simplicity, fewness of parts, and easy and convenient arrangements for the adjustment of its working parts; as its economy will depend on the harmonious action of its reciprocating and revolving mechanism, as well as on the nature of the material and the excellence of the workmanship employed in its construction.

Duplicating the Parts of Steam-Engines.

Duplicating the parts of any class of machines is an advantage, as it insures more uniform proportions in their original construction than could otherwise be obtained, as the term duplication of parts conveys the impression that they are made to standard gauges, and for any number of machines must retain the proportions of the original. While duplication of parts is convenient, and sometimes of great value in cases of emergency, it is rarely so in case of repairs; since, as soon as any journal or bearing is put into use, its dimensions begin to change, the cylinder commences to enlarge and the piston to diminish. This change of shape extends to the piston-rod, and glands of the stuffing-boxes, wrist-pins, crank-pins, rocker-shafts, etc. The eccentric wears flat on two sides, in consequence of the thrust at these points, and the straps wear flat, owing to the push and pull at two points.

Now how can it be expected that a new eccentric will fit the old straps, or the new straps conform to the old eccentric, or that the new piston will prove a good fit for the old cylinder, or the new piston-rod for the worn-out gland? If the crank-shaft becomes worn oval, it will not adjust itself to a new main-bearing made from the original standard; or if the crank-pin becomes worn tapering, owing to the engine being out of line, a box made of the original proportions will not drop into its place and work harmoniously; but, as before stated, in case of emergency, such as break-downs, or where interruption to business would entail great loss, duplicate parts are a tolerably good make-shift, and that is all that can be said in their favor. For this reason, the duplication of parts, which in case of breakage would be most likely to disable a machine, ought to be encouraged, especially in case of marine engines, locomotives running in sections of the country where there are no repair shops, and stationary engines located in isolated places.

Fitting the Cranks of Steam-Engines to their Shafts.

Boring the hole for the shaft in the crank is not so easy a task as the average engineer would suppose. Theoretically, when the hole is bored in the crank, if the boss is faced true, and then bolted to a true face-plate on a lathe, it must be true. But inaccuracy frequently arises from the fact that there are few face-plates which are true, and continue to remain so for any length of time. And even when the boring is as well done as can be expected under the circumstances, the crank is frequently thrown out of line in keying it on the shaft. For this reason, no crank should leave the works where it was made without being tested after having been keyed on.

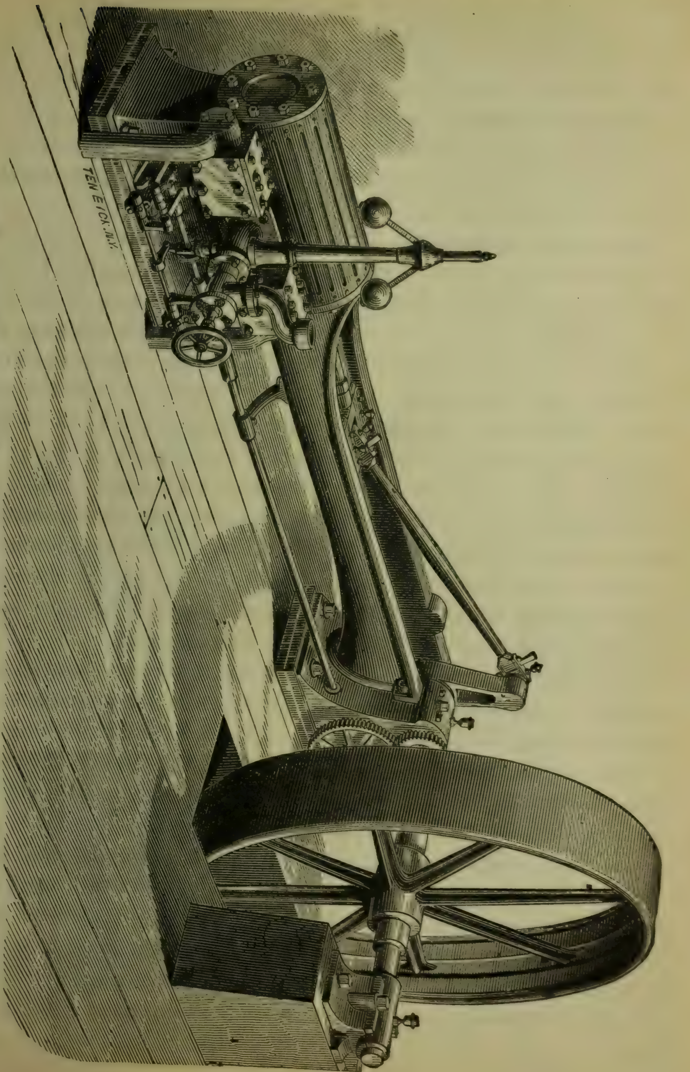
When the crank is in the form of a disc, or wheel, the best plan is to turn it true, first on a mandril, and then so fit it to the shaft, and the key to its seat, that after the keying it will run true; but with the ordinary crank, this cannot be so easily done, as all the surface available for testing its truth is near the centre; in such cases, the main reliance must be placed in fitting the key as well as the crank itself to the shaft. The key should never be finally driven till it has first been frequently partially driven, its points of contact filed or scraped, and it fits perfectly its whole length.

The essential conditions necessary for the production of a well-fitting and durable crank-shaft journal are, good material, a stiff, strong lathe, a skilful machinist, and a sharp, well-tempered, and correctly set tool. The finishing cuts should be light, and, if it cannot be made sufficiently smooth with the tool, it must not be filed, but may be ground and polished smooth by blocks of wood, lead, copper, or some other suitable material fitted to the journal in such a manner that the imperfections left by the turning-tool will be corrected instead of aggravated by the use of a file or end of a stick, as is commonly the case. The polishing powder used should be very fine; emery is considered, by many, objectionable for polishing wearing surfaces, but on good homogeneous material, free from flaws, fine emery may be used without any injurious effects.

The Putnam Machine Company's Automatic Cut-Off Engine.

The opposite cut represents the Putnam Machine Company's high-pressure variable cut-off engine, the frame or bed-plate of which is designed with reference to strength and rigidity, and also to answer either for right or left hand pillow-block bearings. The almost universal reciprocating valve-motion derived from an eccentric is dispensed with, and a rotary motion derived from a gear on the crank-shaft substituted. By means of mitre-gears the motion is communicated to a shaft running parallel with the axis of the cylinder beneath the valves, and carrying cams for lifting the latter. The steam-chests, one at either end of the cylinder, contain each a steam and an exhaust valve of the balance or double-poppet form, having flat faces and seats, and are capable of being removed entire from the steam-chests by simply removing a bonnet or cover on the top of the latter. The valve-stems pass through the necessary stuffing-boxes in the bottom of the steam-chests. The shape and adjustment of the cams for working the valves give them the proper lift, lap, lead, etc. The opening and closing of the valves are very quick, the duration of opening being an interval of rest between the upward and downward motions.

The governor, which is of the ordinary centrifugal form, is driven by bevel-gears from the cam-shaft, thus receiving a positive motion. Below the cam-shaft is a rack-shaft having three arms, the centre one of which is attached to the lifting-rod or spindle of the governor, from which the rack-shaft receives a slight oscillating motion, while those at the ends, which are at right angles with the centre arm, connect with the lifting toes of the steam-valve. The shape of the lower faces of the lifting toes which rest upon the cams is such, that when moved inward towards the cylinder, by the motion of the governor transmitted through the rack-shaft, a curved upward offset is reached by the cam as it revolves, and the valve is lowered so quickly as to have the effect of being actually released and allowed to drop to its seat, while at the same time it is supported by the lifter. The interval between



The Putnam Machine Company's Automatic Cut-Off Engine.

the full lift of the valve and the reaching of the offset by the highest point of the cam determines the point of cut-off, and insures sufficient lift of the valves. The advantage claimed for this arrangement is, that by keeping the valve always supported while open, the danger of slamming is avoided without the necessity of a dash-pot, which, in cases where the valve is tripped or released, is absolutely indispensable.

When these engines are started, and until the speed for which the governor is adjusted is reached, the steam necessarily follows full stroke, as the cut-off is inoperative. But as soon as the regular speed is attained, the motion of the governor thrusts the centre arm of the rack-shaft downward, thereby causing the arms to which the lifting toes are connected to move towards the cylinder. This brings the offsets of the lifting toes nearer to the cams, causing them to drop sooner, thus cutting off the steam at the proper point. In case of the removal of the entire load from the engine, induced by the breaking of a belt, etc., the governor, owing to its positive motion, will effectually check any attempt at "running away," as the offsets on the lifting toes will be thrust so far inward, that the cams will not raise the valves from their seats until the speed is again reduced to the proper point.

It is claimed that, under the above-mentioned circumstances, the engine will not make one full revolution before being completely under the control of the governor. All the sliding and bearing surfaces of the valve-gear of these engines are made of hardened steel, thus preventing the liability of rapid wear, and also requiring very little power to move the valves. The fly-wheels are turned on the face and edges. The shafts, crank-pins, and connecting-rods are made of the best material, and the bearings are ample and well proportioned. The workmanship is excellent, and the finish neat and attractive. In fact, these engines rank among the most simple, durable, and economical in the country. They are manufactured by the Putnam Machine Company, Fitchburg, Mass.

How to put an Engine in Line.

An engine is in line when the axis of the cylinder and the piston-rod are in one and the same straight line in all positions. This line extended should intersect the axis of the engine-shaft, and be at right angles to it. The guides should also be parallel thereto. The shaft must be level, but the centre line of the cylinder may be level, inclined, or vertical, according to the design of the engine.

To "line up" an engine, as it is generally termed, take off the cylinder-head, remove the piston, cross-head, and connecting-rod; then with a centre punch make four (4) marks in the counter-bore at each end of the cylinder, at equal distances apart round the bore. Take a piece of stiff hoop-iron with a hole at one end of it, slip it on to one of the stud-bolts of the back cylinder-head, and secure it firmly with a nut, after which it may be bent in the shape of a crank, one end projecting across the cylinder at its centre, at a sufficient distance from it to admit of convenient and accurate measurement. Next draw a fine line through the cylinder, and attach one end of it to the temporary crank above mentioned, and the other end to a stake driven into the floor at the back end of the bed-plate. Then with a piece of hard wood or stiff wire pointed at each end and equal in length to half the diameter of the cylinder, set the line so that, when one point of the wood or wire is inserted in any one of the centre-punch marks at either end of the cylinder, the other end will feel the line. Next see if this line passes through the centre of the shaft; if so, the cylinder is in line with the shaft; if not, one or the other must be moved, which requires both skill and judgment, since engines differ so much in design and construction. Now turn the engine-shaft round till the crank-pin almost touches the line passing through the centre of the cylinder; then ascertain by measurement whether the line is equidistant from the collars on the crank-pin. Then turn the shaft on the other centre until the crank-pin feels the line. If the measures correspond, the shaft

is in line with the cylinder; if not, they will show which end needs to be moved. The operation may have to be gone over several times before a definite conclusion can be arrived at. The shaft may be levelled by placing a spirit-level on it, if there be room; if not, drop a plumb-line passing through the centre of the crank-pin and shaft; then by placing the crank at both centres and at half-stroke, the line will show whether the shaft is level or not. The guides may be brought into line with the cylinder, by measuring from each end of each guide to the line passing through the centre of the cylinder, and moving them until they are parallel to the line and to each other. To adjust them to the horizontal, a spirit-level may be placed on their top faces. If no level is at hand, a square and plumb-line may be used. Where these accessories are not at hand, a straight-edge placed across them will determine by actual measurement whether they are in line with the centre line of the cylinder or not.

Engines get out of line from the following causes: Faults of design, faults of construction, overwork, the character of the work which they are performing, or from the boss of the crank wearing away the face of the main bearing against which it revolves. To move an engine-shaft and pillow-blocks into line with the centre of the cylinder, screw down the caps of the pillow-blocks firmly on the shaft; then slack up on the bolts that tie down the pillow-blocks to the bed-plate, after which the shaft pillow-blocks and fly-wheel may be moved from the back end by means of a lever or jack-screw, after which they should be firmly tied and the set-screws or wedges readjusted. To move a cylinder, if the connections be short and stiff, remove the bolts which tie it to the bed-plate; then measure from the flange of the cylinder to some fixed object, such as a wall, post, or column; cut a plank or scantling about an inch longer than the actual measurement from the cylinder to the wall, so that when placed against the cylinder it may stand slightly oblique; then by driving on the end of the plank with a sledge or heavy hammer, the cylinder may easily be moved. The holes should then be reamed, and new bolts corresponding to

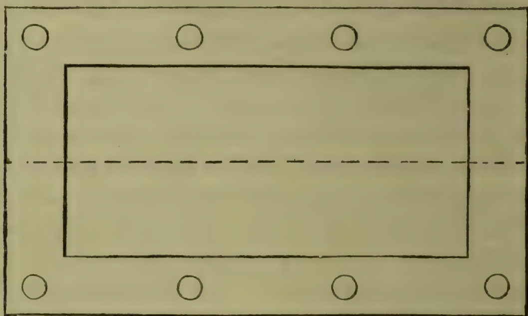
the reamer substituted for the old ones. The cylinders, guides, and pillow-blocks of all engines should be double-pinned to prevent them from getting out of line; and whenever it becomes necessary from wear to move them, the holes may be re-reamed and new pins substituted.

How to Set Up a Stationary Engine.

The first object to determine in setting up steam-engines is to decide definitely the precise point at which the engine is to be located, after which the excavation for the foundation may be made. It should be at least two feet wider and longer than the intended brick- or stone-work, and its depth must depend on the size and weight of the engine and character of the soil. For ordinary sized engines, say from 20 to 40 horse-power, from 3 to 4 feet will suffice, if the soil is dry and firm; but if sandy or swampy, it will require to be sunk deeper. For large engines of from 50 to 100 horse-power it is necessary to find a solid bottom. There are even instances where piles had to be driven to insure a permanent foundation. Too much care cannot be taken in this particular, as any defect in the foundation will materially affect the working of the engine.

Having decided on the location where the engine is intended to stand, line down from the side of the line of shafting, or counter-shaft, if there be any, to the floor, at three or four different places in its length; but if there be no shafting, measure from the side of the building to the centre, at five or six points in its length; then strike a line across all these points. This line will show with sufficient accuracy the line of the building by which the templet may be set up; the latter, as shown in the cut on page 144, should be a fac-simile, or exact counterpart of the bottom of the bed-plate. It may be made of inch pine boards, and set on four props over the excavation, after which it must be squared and levelled with the lines previously taken. The anchor-bolts may now be hung in the templet, and the bricklayers proceed with their work. It

is customary to lay from two to three courses of bricks on the bottom of the foundation before the anchors are reached. These consist of plates of cast-iron or old boiler-plate, generally about a foot square, with a hole sufficiently large for the foundation bolts to slip through; though in some instances the anchors extend entirely across the foundation and take in two bolts each.



The foundation should be widest at the bottom, and slope upwards about 2 inches to the foot, till the level of the floor is reached, after which it may be carried up straight. When finished, it may be an inch wider on each side and end than the bed-plate; after which it should be made perfectly level by means of a coat of good, strong mortar or cement. A parallel piece of pine wood, 1 inch in diameter and from 3 to 4 inches wide, made perfectly straight on both edges, on which a spirit-level may be placed, will answer for levelling the foundation.

After the foundation is level, the bed-plate may be placed on it, either by means of a crane, block and tackle, or skids and blocking, after which it may be tied down and accurately levelled. It is customary, in the case of large engines, to place wedges between the bed-plate and foundation, for the purpose of leaving an interstice between the bottom flange of the bed-plate and the brick work, into which melted sulphur is poured. As sulphur is less

influenced by a change of temperature than any other known mineral, it is of great value as a bedding for heavy steam-engines, and other machinery; besides, when melted, it enters every crevice, and as soon as it is set becomes a permanent fixture. To use it, it is necessary to seal the opening between the bed-plate and brick work, inside and out, with potter's clay, occasionally leaving a gate or "sprue" through which the molten sulphur is poured.

A line should next be accurately drawn through the centre of the cylinder, and attached to some permanent object at the back end of the bed-plate; another line should be drawn at right angles to this through the centre of the main bearing; this latter will give the exact location of the off pillow-block, as the crank-shaft must be exactly at right angles with the horizontal line passing through the centre of the cylinder. The fly-wheel may next be swung into the pit, and the shaft slipped through it and firmly keyed at the right position, after which the pillow-block caps may be screwed down, the front head of the cylinder put on, the cross-head placed in position, the piston slipped in, and the connection between the cross-head and crank-pin made up. Other numerous details might be mentioned, but they never all apply to any individual case, and when any of them present themselves as the work proceeds, the remedy in this case must be prescribed by the erecting engineer. In setting up engines, like setting valves, only general instructions can be given, and it is impossible to lay down any that would apply to each and every case.

How to Reverse an Engine.

Place the crank on the dead-centre and remove the bonnet of the steam-chest; observe the amount of lead or opening that the valve has on the steam end; then loosen the eccentric and turn it round on the main shaft in the direction in which it is intended the engine should run, until the valve has the same amount of lead on the other end. To determine whether the lead is exactly the same at both ends, a small piece of pine wood may be tapered

in the shape of a wedge, and inserted in the port; the marks left on it by the edge of the port and the lip of the valve will show how far it has entered. The engine should then be turned on the other centre for the purpose of equalizing the lead; the crank should also be placed at half-stroke, top and bottom, for the purpose of determining whether the port opening is the same in both positions. When the crank is at half-stroke, the centre of the crank-pin is plumb with the centre of the crank-shaft.

How to Repair Steam-Engines.

It would be reasonable to suppose that any machinist would be capable of repairing steam-engines; and yet, on an examination of numerous cases where repairs have been done by persons calling themselves mechanics, it appears that very few machinists are fit to be trusted to do so. A man to be competent to do repairs must first understand the original character of the engine or machine, and its defects, whether arising from design, inferior material, or workmanship, how an improvement can be made in its working, as well as what would be actually an improvement, before proceeding to make it.

The first step in repairing an engine is to take off the connecting-rod, cross-head, both cylinder-heads, and remove the piston; then pass a line exactly through the centre of the cylinder, and attach it to some fixed object at the back end, to determine if the centre of the crank-pin is in line with the centre of the cylinder. If not, one of them must be moved, and whichever it is will depend on the difficulty to be encountered, and must be determined by the judgment of the party who undertakes the repairs. The cylinder must next be accurately measured at both ends and the centre, for the purpose of determining if it is worn larger in the centre than at either end, or worn oval, as is often the case.

In either case it will be necessary to rebore the cylinder and make it uniform all through. It is next necessary to caliper the cross-head, wrist- and crank-pin, to see if they are worn oval, and

if so, they must be filed round. The guides should then be tested with an accurate parallel piece, to ascertain if they are straight all through; if they are hollow in the middle or at either end, they must be taken down and planed straight. If the piston-rod is badly fluted, it must be put in a lathe, returned, and filed; the rings should be taken off, placed in a lathe-chuck, and faced up on both ends, and if they are cut they should be turned true and smooth. The cross-head should then be measured crosswise to determine whether the guides are too far apart or not; and if so, the holes in the studs which tie them to the bed-plate must be filed oval to bring them to a proper position. If the valve and seat are cut, the valve must be taken off and planed in the opposite direction to its travel. The steam-chest must also be removed, and the valve-seat straightened by filing and scraping, after which the valve may be carefully fitted to it.

The flange of the piston-head and follower-plate should be faced up in a lathe, at the point where they strike the rings, and the latter should be carefully ground and scraped on to them. The piston should next be inserted into the cylinder, set out, and the cross-head slipped on, connected with it, and levelled, so that it may stand parallel with the centre of the axis of the cylinder at all points of the stroke. The connecting-rod boxes should be examined in order to ascertain if they are "brass bound," and if so, they should be filed out. The main pillow-block bearing should receive attention, in order to determine if it is worn oval or loose. In fact, every part should receive attention, because defects that have not been thought of may be revealed as the work progresses. It has been generally heretofore supposed that any one bearing the name of a machinist is competent to repair a steam-engine, which, of course, is a grave error, as thousands of mechanics fully competent to build a machine are totally unfit to repair it.

This arises from the fact, that the repairing of steam-engines and other machinery requires a different class of talent from that necessary to build them. A machinist may be a good hand on either a vice, lathe, or planer; he may be a thorough fitter and a

neat finisher, and yet he may lack that keen observation, that cool, patient, and searching perseverance which are so essential in the party that will become an adept in the repairing of steam-engines and other machinery. It not unfrequently happens, that when everything has been done that was considered absolutely necessary, an engine works badly when started up, which is very discouraging to any one, except those who take a peculiar interest in ferreting out the causes of minor defects which have been overlooked when the more prominent ones were remedied. Almost any one can tell if an engine is badly out of line, the cylinder fluted, or the crank-pins loose or worn oval; but it requires a different kind of talent to determine the different causes for the defective working of steam-engines, and prescribe a remedy for them, as many of them apparently did not exist at the commencement of the work, but cropped out as it progressed. One of the greatest mistakes in the repairs of steam-engines and other machinery, is that those who have them in charge are expected to perform the work in too limited a time. This being impossible, the only resource left is to slight it.

How to Increase the Power of the Steam-Engine.

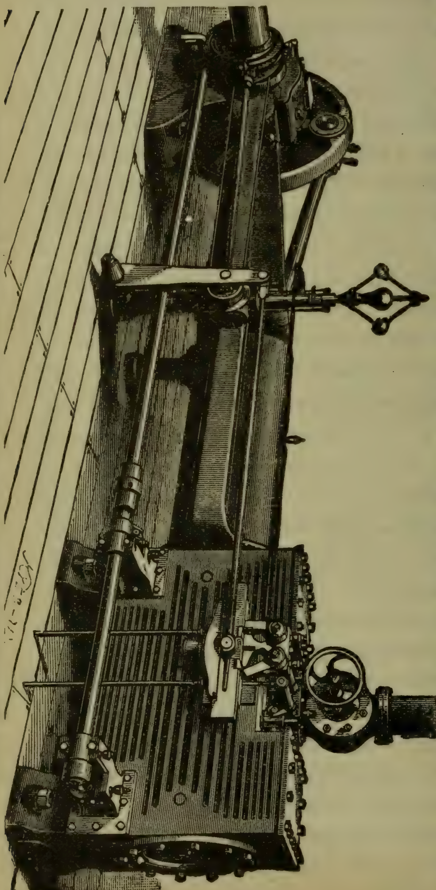
It frequently happens that engines which were originally of sufficient power to do the work of a manufacturing establishment, become unable to do the work, owing to an increase in the business; and while the cost of replacing an engine with one of sufficient power would be a matter of nominal consideration, the time expended in removing and replacing it with a larger one might involve a serious loss to the owner, in case he had large orders for goods to fill at profitable prices. Under such circumstances, the three most practicable ways to remedy the difficulty for the time being would be—*first*, to raise the pressure, providing the boiler is considered safe; *second*, to increase the speed of the engine; *third*, to replace the old cylinder with a new one about two inches larger in diameter, which would of course involve the necessity of a new piston, steam-chest, and valve.

For a moderate increase in power, the last plan would be the most safe and practicable, as the active condition of steam-boilers is not always understood, and without a thorough knowledge on the subject it would be unwise to increase the pressure; nor should any engine be run at a higher speed than it is capable of standing without springing or shaking to pieces. The increase in power that would result from replacing the old cylinder with a new one two inches larger in diameter may be illustrated as follows: Take, for instance, a 10-inch cylinder, which contains 78·54 square inches in area, while a cylinder 12 inches in diameter contains 113·09 square inches, which makes a difference of 34·55 square inches in the piston. Now, if the engine having a 10-inch cylinder was unable to do the work with 60 lbs. pressure per square inch, it would do the work easily with the 12-inch cylinder at the same pressure, as the new cylinder would make a difference of from 5 to 6 horse-power. Measures might be taken, and the new cylinder, piston, and steam-chest prepared and placed in position at a given time, without causing any interruption to the business.

Of course the margin for increasing the size of cylinder for any engine, and using all the other original parts of the engine, is limited, and should never exceed 2 inches; as, to exceed that limit, the other parts would be too light, and become liable to spring. To increase the speed of an engine, it would be necessary to have a new counter-pulley, so that, while the piston velocity is increased, the speed of the shafting may remain the same. An engine will develop more power by increasing its speed, but will use more steam, and as a consequence more fuel will be consumed. The overtaxing of steam-engines and boilers, or any other class of machines, is sure to induce waste either in fuel or wear and tear; but there are circumstances under which manufacturers and steam users find themselves placed, in which it would be impossible to avoid waste. Steam-engines or boilers, or any other class of machines that is too large or too small for the work to be performed, are not economical.

The Improved Greene Automatic Cut-Off Engine.

The illustration represents the "Improved" Greene Automatic Cut-Off Engine, of which the Providence Steam-Engine Company, Providence, R. I., are sole builders.



Front View of the "Improved" Greene Automatic Cut-Off Engine.

The bed-plate is of the girder pattern, symmetrical in appearance, and of ample strength. The slides are cast *separate,* and secured to bed-plate by dowels and bolts. The main journal-boxes are made in four pieces, and provided with set-screws and check-nuts, which permit of convenient and accurate adjustment. The governor is of the Porter pattern, and is driven by a flat belt from the main shaft. The valve-gear is detachable, and is so controlled by the governor that the cutting off may be effected from zero to three-quarters of the entire stroke. The valves are four in number—two steam and two exhaust—and are of the *flat-slide pattern.* The power which moves them

is applied parallel to and in line with their seats, so that they cannot rock or twist—thus obviating the tendency to wear unevenly. The steam-valves when tripped, are shut by the combined action of a weight and *the pressure of the steam on the large valve-stems*, thereby insuring a *quick cut-off*, and the *positive* closing of the port, under all circumstances of speed and pressure. The steam-valves are operated by toes, on the inner ends of two rock-shafts that connect with the valve-stems outside the steam-chest. The outer ends of the rock-shafts are furnished with steel-tipped toes.

There is a sliding-bar carrying tappets which receive a reciprocating rectilinear motion from an eccentric on the main shaft. Below the sliding-bar is a gauge-plate connected with the governor, which receives an up and down motion from a reverse action of the governor balls.

The tappets in the sliding-bar are supported by springs, the lower ends of which rest upon the gauge-plate; the ends of the tappets projecting through the gauge-plate with nuts upon them secured by pins. As the sliding-bar moves, one of the tappets is brought in contact with the inner face of the toe on the rock-lever, causing it to turn on its axis, thereby opening the steam-valve at one end of the cylinder; the other tappet, meanwhile, passes under the rock-lever,—without moving it,—the toe and tappet being so bevelled that the tappet will be forced down against the action of the spring, till it has passed the toe, when the spring causes it to resume its original position, prior to opening the steam-valve at the opposite end of the cylinder upon the return stroke of the bar.

As a result of this motion, the tappet always gives the valves the same lead, and as the bar moves in a straight line, while the toe describes the arc of a circle, the tappet will pass by and liberate the toe, which is brought back to its original position by a weight, and the steam pressure on the large valve-stem, which thus closes the valve and cuts off the steam. The liberation of the toes will take place sooner or later, according to the elevation of the tappet; that is, the lower the tappets are, the sooner the toes will be liberated, and *vice versâ*. By the elevation or depression of the

gauge-plate, the period of *closing* the valves is changed, while the period of *opening* them remains the same. The adjustment of the gauge-plate is effected directly by the governor.

Both the exhaust-valves and seats are convenient of access, and removable from the outside of cylinder. These valves receive their motion from a separate eccentric, thus allowing of easy adjustment, without interference with the steam-valve mechanism. All the connections are on the outside, are few in number, and have ample bearing surfaces, insuring freedom from rapid wear and derangement.

A safety stop-motion is combined with the governor, preventing the admission of steam should the governor-belt run off or break.

The cross-head gibs are directly opposite the centre of pin, thus avoiding any cross strain upon the piston-rod; a lack of attention to this point has been the cause of many serious accidents. The steam-ports are large, thus insuring the full pressure of steam to the point of cut-off. A very desirable feature of this engine, and one that will be appreciated, is the method of connecting the steam-valves with their stems, by which, if water should accumulate in the cylinder, and the engine be started without the usual precautions, the valves will *lift*, giving a *free passage* of the water through the steam-ports. The engine is extremely sensitive to the action of the governor, and is, therefore, particularly adapted to those situations where perfect regulation is required. All parts are well proportioned, made of best material, accurately fitted, and highly finished.

The Dead-Centre.

All reciprocating steam-engines have one dead-centre in each stroke and two in each revolution, and that point is the point at which the steam is exhausted, and the centre of the crank-pin is parallel with the centre of the axis of the cylinder. The centre of the cross-head, in some cases, may be above or below the centre of the cylinder; but by placing a spirit-level on the top or bottom

of the stub-end strap, the dead-centre may be easily found. The experienced engineer or machinist can generally tell by the eye when the crank is at the dead-centre; but to insure accuracy it is always better, in the case of horizontal engines, to try it with a level, and in vertical engines with a plumb-bob and line. The cranks of all engines have to be placed accurately on the centre when the valves are set.

A single reciprocating engine is completely helpless when the crank is at the dead-centre, and would stop there if it was not for the momentum of the balance-wheel. Double reciprocating engines, such as locomotives and marine engines, which have their cranks set at right angles, require no balance-wheel, as they pull each other off the dead-centre, in consequence of one crank being at its full-power point while the other is at the weakest. There are some engines, such as the rotary, which have no dead-centre in their revolution.

The Causes of Knocking in Steam-Engines.

The most frequent causes of knocking in steam-engines are lost motion in the cross-head, wrist- and crank-pin boxes; looseness in the pillow-block or main-bearing boxes; looseness of the piston-rod or follower-plate; the crank-pin or crank-shaft being out of line with the cylinder, or the wrist-pin, crank-pin, or main-bearing journal being worn oval; the slide-valve having too much or not enough lead; the exhaust opening being too soon or too late; the valve being badly proportioned, or the exhaust passage outside of the cylinder being contracted.

Other causes are shoulders being worn in each end of the cylinder, in consequence of the packing-rings not travelling over the counter-bore at each end of the stroke; or shoulders being worn on the guides, resulting from the cross-head shoes not overlapping them when the crank is at the dead-centre; the piston not having sufficient clearance at either end of the cylinder, in consequence of its being altered by taking up the lost motion in

the boxes; there not being sufficient draught in the keys to take up the lost motion in the connecting-rod boxes; the packing being screwed too tight round the piston-rod; excessive cushioning, resulting from the leaky condition of the piston, which allows the steam to occupy the space between the cylinder and piston-head, as the crank approaches the centre, thereby subjecting the engine to an enormous strain, as at this part of the stroke the fly-wheel is travelling very fast and the crank moving very slowly; or lost motion in the connection by which the slide-valve is attached to the rod. Engines out of line frequently knock sideways at the half-stroke, but most generally at the outward or inward, upper or lower dead-centre, as these are the points at which the greatest strain is thrown on the bearings, in consequence of the direction of the connecting-rod having to be reversed. The foregoing causes of knocking in engines constitute the principal ones.

The knocks arising from lost motion in any of the revolving, reciprocating, or vibrating parts of an engine may be located by placing the finger on the part, while the cross-head is being removed back and forth on the guides by the starting-bar; but knocks induced by the valve opening or closing too soon, by a contraction of the exhaust, or by the valve or valves being improperly set, are the most difficult to discover, as they are different from those induced by lost motion, the sound being a dull, heavy *thud*, in many instances causing the engine, building, and even the foundation to vibrate at every stroke. While an intelligent and careful search will in most cases result in successfully locating the knock, some will for a time baffle the most expert engineer. Instances are not uncommon in which weeks have been devoted, engines taken apart and put together again, to find a knock, which, when finally discovered, perhaps turned out to be caused by a loose crank-pin, follower-plate, or key in a fly-wheel. It not unfrequently happens that, after every other means have been resorted to, the indicator has to be applied, in order to determine the precise location of the knock or "thud."

From whatever causes knocking in engines may arise, they are

a nuisance, which sounds harshly not only to the engineer, but to all who have an *ear* for natural mechanics. Nothing, perhaps, makes the intelligent engineer feel so cheap as to be found in charge of an engine that knocks, as lookers-on are not always capable of deciding who is at fault — the engine or the engineer.

The Remedies for Knocking in Steam-Engines.

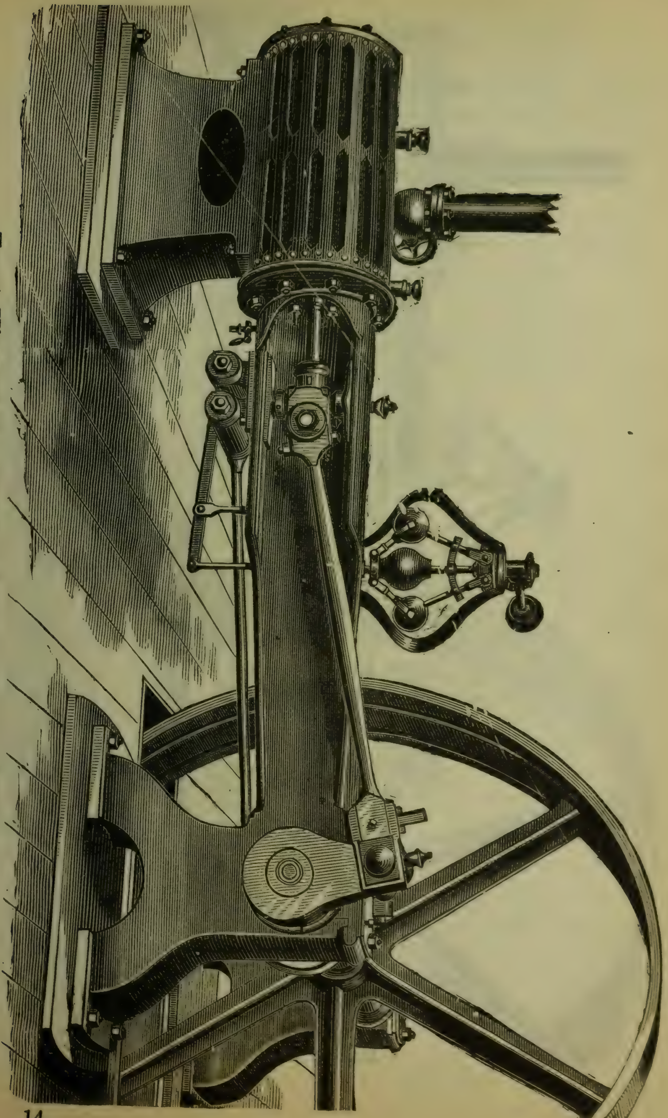
While it may be possible in most cases to locate the knocking in steam-engines, and explain the causes from which they arise, it is hardly possible to prescribe a remedy for all, as, in many instances, it must arise out of and be determined by the circumstances of the individual case. The most practical method of remedying the knocking induced by the crank-pin being out of line, is to place the crank-shaft at right angles with the centre of the cylinder, remove the old crank-pin, rebore the hole so as to bring the centre of the new pin perfectly in line with the axis of the cylinder, and replace the old pin with a new one. The knocking induced by the wrist-pin and crank-pin becoming worn oval, may be remedied by filing them perfectly round; but the knocking caused by the crank-shaft journal being worn out of round is very difficult to remedy; in fact, there is hardly any remedy for it, except to remove the shaft, true it up in a lathe, and refit the boxes, which operation is attended with a good deal of difficulty, more especially when the engine is large.

Knocking in the boxes on the crank-pin and cross-head, or valve-rod, may be remedied by filing out the boxes and readjusting the keys, or by putting a liner behind or in front of the boxes, when there is not sufficient draught in the keys and gibs. Knocking in the steam-chest caused by looseness in the valve connections may be remedied by readjusting the jam-nuts or the yoke. Knocking arising from this cause manifests itself more frequently when steam is shut off from the cylinder, preparatory to stopping the engine, than when the engine is running; the lost motion is taken up in the valve connections by the pressure of the steam on the back of the valve.

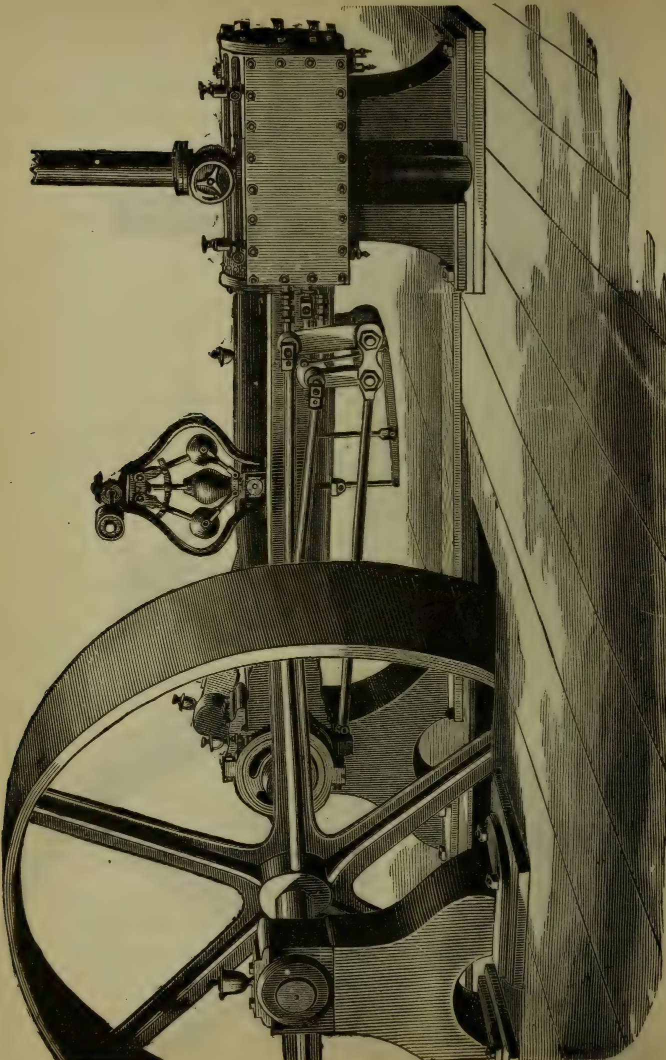
Knocking in the piston is generally caused by the rod becoming loose in the head, and, if it continues for any length of time, it destroys the fit of the rod in the hole. The only practical remedy under such circumstances is to remove the rod, rebore the hole, and bush it or thicken the rod at that point by welding, and fit it to the head after the hole is rebored perfectly true. Knocking in the follower-plate is generally caused by the bolts being too long, or from dirt being allowed to accumulate in the holes, which prevents them from entering sufficiently far to take up the lost motion in the plate, and may be remedied by shortening the follower-bolts, or removing the deposits from the bottoms of the holes, as the case may be.

The knocking caused by shoulders becoming worn in the cylinder at each end can be remedied by reboring the cylinder, and making the counter-bore sufficiently deep that a part of one of the rings will overlap it at each end of the stroke. Knocking caused by shoulders becoming worn on the guides can be remedied by planing the guides and making the gibs or shoes sufficiently long that they will overrun the guides when the crank is at either centre. The knocking induced by any of the foregoing causes is generally a source of great annoyance to the engineer, as any attempt to adjust the boxes on the cross-head or crank-pin, or the piston-packing in the cylinder, generally aggravates the cause of the knocking, as any adjustment of the connecting-rod boxes alters the position of the piston in the cylinder and the cross-head on the guides, and causes them to strike harder against the shoulders.

Knocking caused by the valve or valves being improperly set may be remedied by removing the bonnet of the steam-chest and adjusting the valve, so that it may move uniformly on its seat, thereby giving the same amount of lead at each end of the stroke; then, if the valve is well proportioned, and the connections thoroughly fitted and skilfully adjusted there is no reason why the engine should knock from this cause. But the knocks arising from bad proportion in the valve or steam passages are the most difficult of all to remedy, as they are inherent in the machine.



Front View of the Douglas Automatic Cut-Off Engine.



Back View of the Douglas Automatic Cut-Off Engine.

The Douglas Automatic Cut-Off Engine.

The cuts on pages 157, 158, represent the Douglas Automatic Cut-Off Engine. It will be noticed that the bed-plate is of the girder-frame pattern, which is faced up to receive the cylinder at one end and the main pillow-block bearing at the other. The cylinder rests on a tapering pedestal, while the back end of the bed-plate and crank-shaft bearing is supported by a double leg, which is cast solid with the bed-plate. The pillow-blocks at the cylinder-base are placed on the under side, and are situated at equal distances from the centre, which facilitates the setting up of the engine or placing it in line, as all that is necessary is to level the foundation stone and place the engine on it. The cross-head guides are bored out cylindrical, and on line with the centre of the cylinder, which obviates the liability of the engine getting out of line.

The main steam-valve serves both for induction and exhaust. The exhaust passes through its centre to the exhaust-port at the centre of the cylinder. It receives its motion from an eccentric, through the intervention of a rocker-arm and small take-up connections from the rocker-arm to the valve-rod. The two cut-off valves are flat, and slide on the top of the main valve. They receive their motion from an extra eccentric and rocker-arm. On this rocker-arm is a disc, pivoted on its centre. At equal distances from the pivot-pin, in opposite directions, are two wrist-pins, to which the cut-off valve on the frame end is attached by a take-up connection and spade-handle joints to the lower pin of the disc, while the steel rod passing through the sleeve to move the other cut-off valve is attached to the other pin on the disc. The lever and connection attachments from the governor to the rocker and disc rotate either way, separating the cut-off valves or drawing them nearer together, cutting off the steam earlier or later in the stroke, to accommodate a varying load and pressure.

The governor is very powerful, sensitive, and positive in its action, and can be driven by either belt or gearing. Should the belt

shrink or slip off, the engine would continue to run the same as before it broke, as there would be no power to change the valves, since the centrifugal force has only the clutch-back and the centre-weight to lift. The driving power of the governor, when the clutches are in contact, acting on a clutch attached directly to the top of the screw, turns it up, and, acting on a clutch attached to the reversed gear, turns it down. It turns the screw up or down out of clutch before the governor can make a revolution.

The pillow-block boxes are lined with Babbit metal, and are provided with wedge- and draw-screws for the purpose of taking up the wear and lost motion. The wrist- and crank-pins, valve- and piston-rods, are made of steel well proportioned and well fitted. The fly-wheels are turned off on the face and sides and are accurately balanced. The Douglas engines are in very general use in the Western States and Territories, and wherever used their reputation for efficiency, durability, and economy has added to their credit.

Technical Terms Applied to Different Parts of Steam-Engines.

Bonnet. — This term is applied to the covers of the steam-chest.

Brasses. — This term is understood to apply to the wrist- and crank-pin, or connecting-rod boxes; but it is used in connection with other arrangements.

Counterbore.—A term applied to recesses in the ends of steam-cylinders in the clearance space, over which the piston-rings partly travel. The object of the counterbore is to prevent shoulders being formed at each end of the cylinder, which would induce knocking in the engine when any changes are made in the connecting-rod brasses.

Jam-nuts. — A term applied to the nuts which lock the adjusting-screws in the piston- and valve-gear of steam-engines; but jam-nuts and lock-nuts are used for many other purposes in connection with the steam-engine.

Pipe-swivel.—A long nut containing a right- and left-hand thread. It is used for adjusting the valve-gear of steam-engines, particularly those of the Corliss type; but the pipe-swivel is used for many other purposes than this.

Trunk.—A term applied to the hollow tube connected with the pistons of trunk engines in which the connecting-rod oscillates. The term is just as applicable to certain other parts of machinery and arrangements as to the steam-engine.

Trunnions.—A term applied to the gudgeons on which the cylinders of oscillating engines vibrate; but it may be, and often is, applied to other machinery as well as oscillating engines.

Terms Formerly Applied to Different Parts of Steam-Engines, but which have become Obsolete.

Gab-lever.—A term formerly applied to an arrangement used for lifting and lowering the eccentric-hook off and on the rocker-pin.

Pitman.—A term applied to the crank-pins of steam-engines in early times.

Plug-tree.—A primitive valve-gear which superseded the scoggin.

Radius-bar.—A term applied to the connecting-rods of engines in the early days of steam engineering.

Scoggin.—This name was given by the boy Potter to the arrangement he invented for opening and closing the valves of steam-engines.

Shackle-bar.—This term was used to denote the connecting-rod of steam-engines at a period when they were generally made of wood, and strapped with iron at both ends.

Spider.—The primitive name for piston-heads of steam-engines.

Questions:

THE ANSWERS TO WHICH WILL BE FOUND IN THE TEXT.

Into what two classes are steam-engines divided, regardless of design, general arrangement, etc.?

Into what two classes are steam-engines in general sub-divided?

Explain the difference between condensing and non-condensing engines, their advantages and disadvantages, and their difference in useful results.

What are the advantages of compound over simple engines, and *vice versâ?*

State the formulæ for estimating the power of each class of engines.

Explain the difference between automatic cut-off and throttling engines, and the advantages of the one over the other.

Explain the advantages and disadvantages of the various cut-offs employed on stationary, marine, and locomotive engines.

What are the most valuable features in any steam-engine?

What advantages are derived from duplicating the parts of steam-engines?

How would you proceed to fit the crank of a steam-engine on its shaft?

How would you proceed to set up a stationary engine?

How would you proceed to repair a steam-engine?

What is the meaning of the term "dead-centre"? How would you proceed to find it?

Explain the general causes of knocking in steam-engines, and the remedies for the same.

PART THIRD.

Bed-Plates and Housings.

The bed-plate is that part of a steam-engine which forms the connection between the cylinder and the main pillow-block or crank-shaft, and, in many instances, constitutes the support on which they rest. They embrace a great variety of shapes and forms, such as the box side bed-plate, girder-frame, etc., which were all doubtless designed to meet some peculiar requirement, and for each of which special advantages are claimed, the girder-frame being in most favor with modern engineers. This is in part due to the fact, that the necessary rigidity can be obtained with less metal than in any other form, and that the strength can be more equally distributed in the line of the strain, and above and below it. Bed-plates are subjected to transverse in addition to tensile and compression strains.

In designing a girder-frame, if it is to be supported only at the ends, due allowance must be made for transverse strains due to the thrust of the connecting-rod. The amount of this strain may be found by dividing the greatest pressure to which the piston may be subjected at mid-stroke by the quotient obtained by dividing the connecting-rod by the crank. Thus, suppose the area of the piston is 200 sq. inches, and it is desired to give ample strength for, say 80 lbs. of steam at mid-stroke; $200 \times 80 = 16{,}000$ lbs., the force on the piston. Then suppose the quotient obtained by dividing the connecting-rod by the crank is 5; $16{,}000 \div 5 = 3200$ lbs., the pressure on the slides at mid-stroke. When the engine runs over, that is, when the top of the fly-wheel runs from the cylinder, the weight of the cross-head, and half the weight of the connecting- and piston-rods, must be added to this, and deducted when the motion is in the opposite direction.

When an engine runs under, a support under the frame at the slides (supposing the frame to be of the girder type) would not compensate for weakness of the frame, as the thrust of the connecting-rod being upwards, the upper slide would give, however securely the lower one might be supported. The term housing is applied to the upright frames of both land and marine engines.

The Housing.— This term is applied to the upright frame of vertical engines on which the cylinder rests, and which, at its base, contains the main pillow-block bearings.

Steam-Cylinders.

The cylinder is one of the most important as well as the most expensive parts of a steam-engine; it must be made of iron possessing the qualities of hardness and toughness, be moulded and cast with great care, and bored with great accuracy. Cylinders, from the moment they are put into use, have a tendency to wear oblong, also to wear larger in some places than others. This involves the necessity of reboring them, which is one of the largest items of expense incurred in the repairs of a steam-engine.

There are certain peculiarities connected with the wear of steam-cylinders upon which engineers have hitherto been unable to agree, among which is, why the cylinders of different engines of the same size, design, and manufacture, and under the same conditions, wear in opposite directions. The cylinders of some horizontal engines wear more on the lower than on the upper side, while others of the same size and build wear more on the sides opposite the ports, and others on the sides next the ports. Nor is it always the largest cylinders and heaviest pistons that wear most on the lower side of the cylinder. The same peculiarities hold good in relation to vertical engines. On some lines of ocean steamers, where four or five of the engines were built by the same manufacturing firm, and whose design, quality of material, character of workmanship were intended to be as much alike in every respect as it was possible to make them, it was found on exami-

nation that the cylinders of all were worn oblong — some in the middle, others at both ends, and others still at only one end. It is a general impression among engineers, that the cylinders of very large horizontal engines are more liable to wear oblong than those of vertical engines of the same bore; but experience and observation have proved this to be a mistaken idea. A distinguished American mechanic, who has had more experience in boring out the cylinders of large stationary, locomotive, and marine engines, within the past ten years than any other party on this continent, asserts that there is no accounting for the manner in which steam-cylinders wear, and that in numerous instances he found the cylinders of the engines of ocean steamers worn oblong, the wear being as often on the sides next the ports as on those opposite. He also observed in horizontal engines, with cylinders 36 inches in diameter, that the wear on the bottom was hardly perceptible, while it was sufficiently apparent, on either one side or the other, to involve the necessity of reboring.

This is a subject worthy of study and investigation, as on it depends a good deal of the economy of the steam-engine. Most engineers would be inclined to think that such freaks were due to a want of perfect alignment, as, with the piston, cross-head, crank-pin, perfectly true with the centre-line of the cylinder, and with each other, it is difficult to see why the piston should press in any direction except that caused by gravity; but most experienced engineers are aware that engines that are supposed to be perfectly in line are not actually so, and a very little inaccuracy in the alignment of the slides, or in the cross-head guides, may suffice to press the piston out of centre. Even this may be aggravated by any unequal thickness of packing in the stuffing-box around the piston-rod.

Rule for finding the proper thickness for steam-cylinders.

Divide the diameter of the cylinder plus 2 by 16, and deduct a $\frac{1}{100}$th part of the diameter from the quotient; the remainder will be the proper thickness.

Rule for finding the required diameter of cylinder for an

engine of any given horse-power, the travel of piston and available pressure being given.

Multiply 33,000 by the number of horse-power; multiply the travel of piston in feet per minute by the available pressure in the cylinder. *Divide* the first product by the second; divide this quotient by the decimal ·7854. The square root of the last quotient will be the required diameter of cylinder.

Rule for finding the cubic contents of a steam-cylinder.

Multiply the area of cylinder in inches by the length of the stroke in inches, and *divide* this product by 1728. The quotient will be the number of cubic feet.

TABLE

SHOWING THE PROPER THICKNESS FOR STEAM-CYLINDERS FROM 6 TO 90 INCHES.

Diameter of Cylinder.	Thickness.	Diameter of Cylinder.	Thickness.	Diameter of Cylinder.	Thickness.	Diameter of Cylinder.	Thickness.
6 in.	·440	28 in.	1·595	50 in.	2·750	72 in.	3·905
8 "	·545	30 "	1·700	52 "	2·855	74 "	4·010
10 "	·650	32 "	1·805	54 "	2·960	76 "	4·115
12 "	·755	34 "	1·910	56 "	3·065	78 "	4·220
14 "	·860	36 "	2·015	58 "	3·170	80 "	4·325
16 "	·965	38 "	2·120	60 "	3·275	82 "	4·430
18 "	1·070	40 "	2·225	62 "	3·380	84 "	4·535
20 "	1·175	42 "	2·330	64 "	3·485	86 "	4·640
22 "	1·280	44 "	2·435	66 "	3·590	88 "	4·745
24 "	1·385	46 "	2·540	68 "	3·695	90 "	4·850
26 "	1·490	48 "	2·645	70 "	3·800		

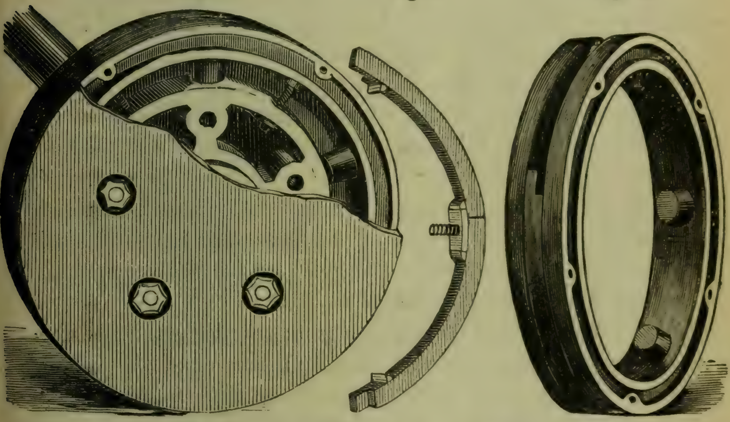
Rule for finding the quantity of steam any engine will use at each stroke of the piston.

Multiply six times the area of the cylinder by $\frac{1}{6}$ the stroke, and *divide* by 1728; the quotient is the cubic contents of the cylinder in feet. *Divide* this quotient by the cut-off $\frac{1}{2}$, $\frac{1}{4}$, or $\frac{1}{8}$, as the case may be; the result will be the quantity of steam used at each stroke of the piston.

Cylinder-head bolts.—There does not appear to be any uni-

versal rule among steam-engine builders for proportioning the strength of cylinder-head bolts. In most of the prominent locomotive works in this country, eleven $\frac{7}{8}$ bolts for an 18-inch cylinder are used; which practice is based on the assumption that 150 lbs. of steam pressure per square inch is the maximum strain to which the area of the head can safely be subjected. Taking 5800 as the sectional area of each bolt, and dividing by the total pressure of steam per square inch against the cylinder-head, we get the area of all the bolts required. This quotient, if divided by the area of one bolt, will give the whole number of bolts necessary.

Fig. 1. Fig. 3. Fig. 2.



The Babbit & Harris Steam-Piston.

Steam-Pistons.

The piston is one of the most important adjuncts of the steam-engine; all the other parts are subsidiary to it. No part of the steam-engine, since its advent, has proved a greater source of annoyance to the engineer, and anxiety and waste to the steam user. It is well known that only about 10 per cent. of the energy stored up in good fuel is utilized in the best class of steam-engines; this

being a fact, however economically steam may be generated in the boiler; unless the piston is steam-tight and capable of resisting the strains to which it is subjected, very little of the work it should perform will be realized.

There are strong reasons why every portion of an engine should be made as light as is consistent with strength; but this is especially the case in the piston, from the rapidity of its reciprocating motion and the strains induced by the momentum on the crank-pin and other parts of the mechanism; consequently, the essential requirements of a good piston are strength, lightness, simplicity, durability, and convenient arrangement for easy and accurate adjustment. Though the U. S. Patent-Office is literally crowded with arrangements which are claimed to be improvements on all former devices, it is asserted by intelligent engineers that a good piston is as much of a necessity as it was in the days of Watt. Nor has it ever been definitely settled which of the steam-pistons now in use is best suited to all classes of engines; nor is it at all likely that any one piston will ever be able to establish its superiority under all circumstances. It may be said of steam-pistons, as of steam-engine governors, while they behave well in the majority of cases, there are circumstances under which the very best of them utterly fail to give satisfaction.

The depth of the piston-rings, in good practice, should be about $\frac{1}{4}$ the diameter of the cylinder, and the thickness of the follower-plate the same as that of the cylinder; so that the whole thickness of the piston will be $\frac{1}{4}$ the diameter of the cylinder plus twice its thickness, as obtained by the foregoing rule. The diameter of the piston-rod should be from $\frac{1}{5}$ to $\frac{1}{6}$ that of the cylinder for high-pressure engines, and $\frac{1}{7}$ for condensing engines.

The cuts on page 167 show the Babbit & Harris Piston, which is in very general use, and is said to be very serviceable. No. 1 represents the packing in its place; No. 2 shows the junk-ring, with two sections of packing out; No. 3, the said two sections.

The inner ring of steam-piston packings, against which the springs press, is termed the junk-ring.

Piston-Rods.

The diameter of piston-rods varies with different builders, the range being between $\frac{1}{6}$ and $\frac{1}{10}$ the diameter of the cylinder, according to their length and probable maximum pressure. The high-pressure piston-rods of the American line of steamships are about $\frac{1}{7}$ the diameter of the cylinders, and the low-pressure about $\frac{1}{10}$. The piston-rod of the Corliss Centennial Engine was about $\frac{1}{6}$ the diameter of the cylinder. A rod $\frac{1}{10}$ the diameter would be $\frac{1}{100}$ the area of piston; and if 100 lbs. of steam were acting on the piston, the strain would be 10,000 lbs. per square-inch section of rod, which is about $\frac{1}{5}$ the breaking strength of good iron.

But the strain on a piston-rod is alternately tensile and compressive. Such a size would evidently do for such a pressure, though it might not break so long as it was not subjected to any undue strains from accidental causes, such as water in the cylinder, etc. On the other hand, the largest size in use — $\frac{1}{6}$ the diameter of the cylinder — would be $\frac{1}{36}$ the area, on which the strain due to 100 lbs. of steam would be 3600 lbs. per square-inch section, which is fairly within the limits of perfect safety. But the pressure on the piston is not the main consideration in determining the size of the rod, as accidental strains, to which it is liable to be subjected, must be adequately provided for. Some of these strains bear no relation to the steam pressure, so that the diameter of the piston should be made the main factor in determining the size of the rod. Bourne's rule is to *multiply* the diameter of the cylinder in inches by the square root of the pressure on the piston in pounds per square inch, and *divide* the product by 50. The quotient is the size of the piston-rod.

Piston-rods may be smaller in diameter than the foregoing, if made of steel, and if they possess sufficient rigidity and strength to resist all strains to which they may be exposed, and at the same time induce less friction, do more service, with less liability to flute or require returning, while the difference in first cost would be very trifling, and that of fitting about the same.

TABLE

OF UNITS OF HORSE-POWER FOR DIFFERENT PISTON SPEEDS.

The following table will supply any units of horse-power, besides those already given, for any other velocity of piston by multiplication or division. For example, a piston of 12 inches diameter, at 400 feet per minute, gives 1·366 horse-power for every pound average pressure on each square inch, and will give one-half or double this amount at speeds of 200 or 800 feet a minute.

INDICATED HORSE-POWER FOR EACH POUND AVERAGE PRESSURE PER SQUARE INCH, WITH DIFFERENT DIAMETERS AND SPEEDS OF PISTON.

Diameter of Cylinder.	SPEED OF PISTON IN FEET PER MINUTE.							
	240	300	350	400	450	500	550	600
Inches.								
4	·091	·114	·133	·152	·171	·19	·209	·228
4½	·115	·144	·168	·192	·216	·24	·264	·288
5	·144	·18	·21	·24	·27	·30	·33	·36
5½	·173	·216	·252	·288	·324	·36	·396	·432
6	·205	·256	·299	·342	·385	·428	:471	·513
6½	·245	·307	·391	·409	·461	·512	·563	·614
7	·279	·348	·408	·466	·524	·583	·641	·699
7½	·321	·401	·468	·534	·602	·669	·735	·802
8	·365	·456	·532	·608	·685	·761	·837	·912
8½	·413	·516	·602	·688	·774	·86	·946	1·032
9	·462	·577	·674	·770	·866	·963	1·059	1·154
9½	·515	·644	·751	·859	·966	1·074	1·181	1·288
10	·571	·714	·833	·952	1·071	1·390	1·309	1·428
10½	·63	·787	·919	1·050	1·181	1·313	1·444	1·575
11	·691	·864	1·008	1·152	1·296	1·44	1·584	1·728
11½	·754	·943	1·1	1·257	1·414	1·572	1·729	1·886
12	·820	1·025	1·195	1·366	1·540	1·708	1·880	2·050
13	·964	1·206	1·407	1·608	1·809	2·01	2·211	2·412
14	1·119	1·398	1·631	1·864	2·097	2·331	2·564	2·797
15	1·285	1·606	1·873	2·131	2·409	2·677	2·945	3·212
16	1·461	1·827	2·131	2·436	2·741	3·045	3·349	3·654
17	1·643	2·054	2·396	2·739	3·081	3·424	3·766	4·108
18	1·849	2·312	2·697	3·083	3·468	3·854	4·239	4·624
19	2·061	2·577	3·006	3·436	3·865	4·295	4·724	5·154
20	2·292	2·855	3·331	3·807	4·265	4·759	5·234	5·731
21	2·518	3·148	3·672	4·197	4·722	5·247	5·771	6·296
22	2·764	3·455	4·031	4·607	5·183	5·759	6·334	6·911

TABLE — (*Continued.*)

INDICATED HORSE-POWER FOR EACH POUND AVERAGE PRESSURE PER SQUARE INCH, WITH DIFFERENT DIAMETERS AND SPEEDS OF PISTON.

Diameter of Cylinder.	SPEED OF PISTON IN FEET PER MINUTE.							
	240	300	350	400	450	500	550	600
Inches.								
23	3·021	3·776	4·405	5·035	5·664	6·294	6·923	7·552
24	3·289	4·111	4·797	5·482	6·167	6·853	7·538	8·223
25	3·569	4·461	5·105	5·948	6·692	7·436	8·179	8·923
26	3·861	4·826	5·630	6·435	7·239	8·044	8·848	9·652
27	4·159	5·199	6·066	6·932	7·799	8·666	9·532	10·399
28	4·477	5·596	6·529	7·462	8·395	9·328	10·261	11·193
29	4·805	6·006	7·007	8·008	9·009	10·01	11·011	12·012
30	5·141	6·426	7·497	8·568	9·639	10·71	11·781	12·852
31	5·486	6·865	8·001	9·144	10·287	11·43	12·573	13·716
32	5·846	7·308	8·526	9·744	10·962	12·18	13·398	14·616
33	6·216	7·770	9·065	10·360	11·655	12·959	14·245	15·54
34	6·59	8·238	9·611	10·984	12·357	13·73	15·103	16·476
35	6·993	8·742	10·199	11·656	13·113	14·57	16·027	17·484
36	7·401	9·252	10·794	12·336	13·878	15·42	16·962	18·504
37	7·819	9·774	11·403	13·032	14·861	16·29	17·919	19·548
38	8·246	10·308	12·026	13·744	15·462	17·18	18·898	20·616
39	8·648	10·86	12·67	14·48	16·29	18·1	19·91	21·62
40	9·139	11·424	13·328	15·232	17·136	19·04	20·944	22·848
41	9·604	12·006	14·007	16·008	18·009	20·00	22·011	24·012
42	10·065	12·594	14·693	16·792	18·901	20·99	23·089	25·188
43	10·56	13·20	15·4	17·6	19·8	22·0	24·2	26·4
44	11·046	13·818	16·121	18·424	20·727	23·03	25·333	27·636
45	11·563	14·454	16·863	19·272	21·681	24·09	26·399	28·908
46	12·086	15·128	17·626	20·144	22·662	25·18	27·698	30·216
47	12·614	15·768	18·396	21·024	23·652	26·28	28·908	31·536
48	12·846	16·446	19·187	21·928	24·669	27·41	30·151	32·152
49	12·913	17·142	19·999	22·856	25·713	28·57	31·427	34·284
50	14·28	17·85	20·825	23·8	26·775	29·75	32·725	35·7
51	14·832	18·54	21·665	24·76	27·855	30·95	34·045	37·08
52	15·437	19·296	22·512	25·728	28·944	32·16	35·376	38·592
53	16·041	20·052	23·394	26·736	30·078	33·42	36·762	40·104
54	16·656	20·82	24·29	27·76	31·23	34·7	38·17	41·64
55	17·275	21·594	25·193	28·792	32·391	35·99	39·589	43·188
56	17·909	22·386	26·117	29·848	33·579	37·31	41·041	44·772
57	18·557	23·196	27·062	30·928	34·794	38·66	42·526	46·392
58	19·214	24·018	28·021	32·024	36·027	40·03	44·033	48·036
59	19·902	24·852	28·994	33·136	37·278	41·42	45·562	49·704
60	20·558	25·698	29·981	34·264	38·547	42·83	47·113	51·396

TABLE

SHOWING LENGTH OF STROKE AND NUMBER OF REVOLUTIONS FOR DIFFERENT PISTON SPEEDS IN FEET PER MINUTE.

STROKE.	SPEED OF PISTON IN FEET PER MINUTE.														
	Ft. 200	210	220	225	230	240	250	260	270	280	290	300	320	340	350
2 in.	Rev. 600	630	660	675	690	720	750	780	810	840	870	900	960	1020	1050
3 "	400	420	440	450	460	480	500	520	540	560	580	600	640	680	700
4 "	300	315	330	337	345	360	375	390	405	420	435	450	480	510	525
5 "	240	252	264	270	276	288	300	312	324	336	348	360	384	408	420
6 "	200	210	220	225	230	240	250	260	270	280	290	300	320	340	350
7 "	170	180	188	193	197	206	214	223	231	240	248	257	274	291	300
8 "	150	157	165	169	172	180	187	195	202	210	217	225	240	255	262
9 "	133	140	147	150	153	160	166	173	180	187	193	200	213	227	233
10 "	120	126	132	135	138	144	150	156	162	168	174	180	192	204	210
11 "	109	114	120	123	125	131	136	142	147	153	158	164	174	185	191
1 ft. 0 "	100	105	110	112	115	120	125	130	135	140	145	150	160	170	175
1 " 1 "	92	97	101	104	106	111	115	120	125	129	134	138	148	157	161
1 " 2 "	86	90	94	96	98	103	107	111	116	120	124	128	137	146	150
1 " 3 "	80	84	88	90	92	96	100	104	108	112	116	120	128	136	140
1 " 4 "	75	79	82	84	87	90	94	97	101	105	109	112	120	127	131
1 " 5 "	70	74	78	79	81	85	88	92	95	99	102	106	113	120	123

TABLE

(By permission, from Auchincloss' "Link and Valve Motions.")

SHOWING LENGTH OF STROKE AND NO. OF REVOLUTIONS FOR DIFFERENT PISTON SPEEDS IN FEET PER MINUTE.

STROKE.	SPEED OF PISTON IN FEET PER MINUTE.														
	Ft. 200	210	220	225	230	240	250	260	270	280	290	300	320	340	350
1 ft. 6 in.	Rev. 67	70	73	75	76	80	83	86	90	93	97	100	106	113	116
1 " 8 "	60	63	66	68	70	72	75	78	81	84	87	90	96	100	105
1 " 10 "	55	57	60	61	63	66	68	71	74	76	79	82	88	93	96
2 " 0 "	50	52	55	56	57	60	63	65	67	70	72	75	80	85	87
2 " 3 "	44	47	49	50	51	53	55	58	60	62	64	66	72	76	78
2 " 6 "	40	42	44	45	46	48	50	52	54	56	58	60	64	68	70
2 " 9 "	36	38	40	41	42	43	45	47	49	51	53	55	58	62	64
3 " 0 "	33	35	36	37	38	40	42	43	45	47	48	50	53	56	58
3 " 3 "	31	32	33	34	35	37	38	40	41	43	44	46	50	52	54
3 " 6 "	29	30	31	32	33	34	36	37	38	40	41	43	46	48	50
3 " 9 "	27	28	29	30	31	32	33	34	36	37	39	40	43	45	47
4 " 0 "	25	26	27	28	29	30	31	32	34	35	36	38	40	42	44
4 " 3 "	23	24	25	26	27	28	29	30	32	33	34	35	38	40	41
4 " 6 "	22	23	24	25	26	27	28	29	30	31	32	33	35	38	39
4 " 9 "	21	22	23	23	24	25	26	27	28	29	30	31	33	36	37
5 " 0 "	20	21	22	22	23	24	25	26	27	28	29	30	32	34	35

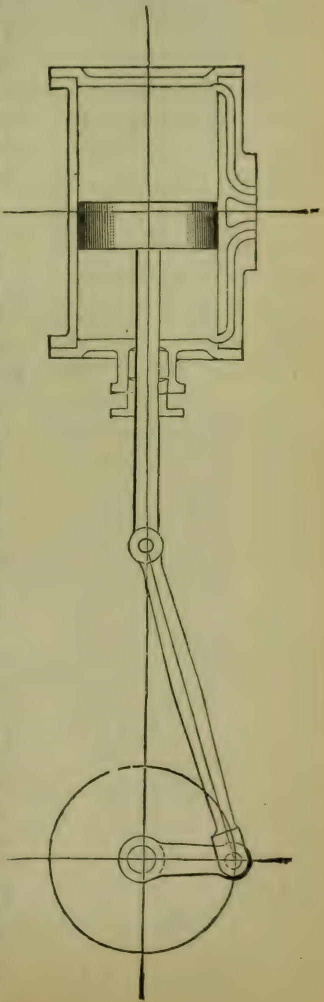
TABLE

SHOWING LENGTH OF STROKE AND NUMBER OF REVOLUTIONS FOR DIFFERENT PISTON SPEEDS IN FEET PER MINUTE.

Stroke.	Speed of Piston in Feet per Minute.														
	Ft. 360	370	380	390	400	410	420	430	440	450	460	470	480	490	500
1 ft. 6 in.	Rev. 120	123	126	130	133	136	140	143	146	150	153	156	159	163	166
1 " 8 "	108	111	114	117	120	123	126	129	132	135	138	141	144	147	150
1 " 10 "	98	101	103	104	109	112	114	117	120	122	125	128	131	133	136
2 " 0 "	90	92	95	97	100	102	105	107	110	112	114	117	120	122	125
2 " 3 "	80	82	84	86	89	91	93	95	97	100	102	104	106	108	111
2 " 6 "	72	74	76	78	80	82	84	86	88	90	92	94	96	98	100
2 " 9 "	65	67	70	71	72	74	76	78	80	81	83	85	87	89	91
3 " 0 "	60	61	63	65	66	68	70	71	73	75	76	78	80	81	83
3 " 3 "	55	56	58	60	61	63	64	66	67	69	70	72	74	75	76
3 " 6 "	51	52	54	55	57	58	60	61	62	64	65	67	68	70	71
3 " 9 "	48	49	50	52	53	54	56	57	58	60	61	62	64	65	66
4 " 0 "	44	46	47	48	50	51	52	53	55	56	57	58	60	61	62
4 " 3 "	42	43	44	45	47	48	49	50	51	53	54	55	56	57	58
4 " 6 "	40	41	42	43	44	45	46	47	48	50	51	52	53	54	55
4 " 9 "	38	39	40	41	42	43	44	45	46	47	48	49	51	52	53
5 " 0 "	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50

Piston, Connecting-Rod, and Crank Connections.

An idea very generally prevails among engineers that the crank of a steam-engine travels faster at one part of the stroke than at the other. This is evidently a mistake. The crank travels at a uniform speed throughout its revolution, but the piston travels farther to make one-half its stroke than the other. If the connecting-rod were indefinitely long, or a slotted yoke were substituted for it, the movement of the piston would be determined by the crank alone; its points of mid-travel would correspond exactly with the corresponding points in the travel of the crank, and the piston would occupy the same position at the first and last half of each stroke. But in consequence of the distorting action of the connecting-rod, the piston travels farther during the half of each stroke farthest from the crank, and consequently, when the crank is at its point of mid-travel, that is, when it is perpendicular to the axial line of the cylinder, the piston is nearer the crank than its point of mid-travel by an amount which varies inversely with the length of the connecting-rod, and which is equal to the difference between the base and the hypothenuse of the right-angled triangle formed by the



connecting-rod, crank, and the included portion of the line. Now the square of the hypothenuse of a right-angled triangle is equal to the sum of the squares of the other two sides.

The crank of a steam-engine moves six times as far while the piston is travelling the first inch of the stroke as while it is making the middle inch; a little over twice as far while the piston is moving the second inch; a trifle over $1\frac{1}{2}$ times as far while the piston moves the third inch; and less than $1\frac{1}{2}$ times as far while the piston is making the fourth inch. The crank also travels less when the piston is making the last inch of the stroke than it does while it is making the first. Another fact, not generally recognized by inexperienced persons, is that the crank of a steam-engine at certain points travels a considerable distance, while the cross-head has a motion which is hardly perceptible.

Rule for finding the distance the piston is ahead of a central position in the cylinder on the forward stroke, and also the distance which it lags behind on the backward stroke.

Subtract the square of the length of the crank from the square of the length of the connecting-rod; find the square root of the difference or remainder, and subtract it from the length of the connecting-rod. The remainder will be the variation of the piston from a central position when the crank is at right angles to the centre line of the engine.

Example.—Length of crank, 12 in.

Length of connecting-rod, 72 "

$$
\begin{aligned}
\text{Then } 72^2 &= 5184 \text{ in.} \\
12^2 &= \underline{\ \ 144} \text{ "} \\
\text{Difference} &= 5040 \text{ "}
\end{aligned}
$$

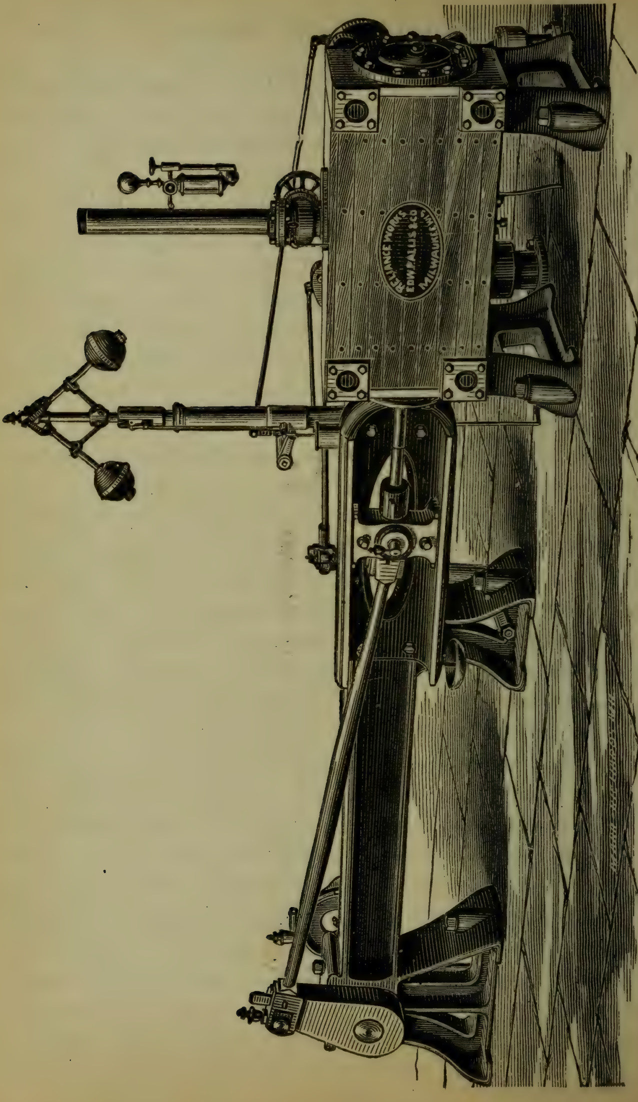
$$\sqrt{5040} = 70{\cdot}992 \text{ in.; and}$$

$$
\begin{array}{l}
72 \\
\underline{70{\cdot}992} \\
1{\cdot}008, \text{ which is the variation in inches.}
\end{array}
$$

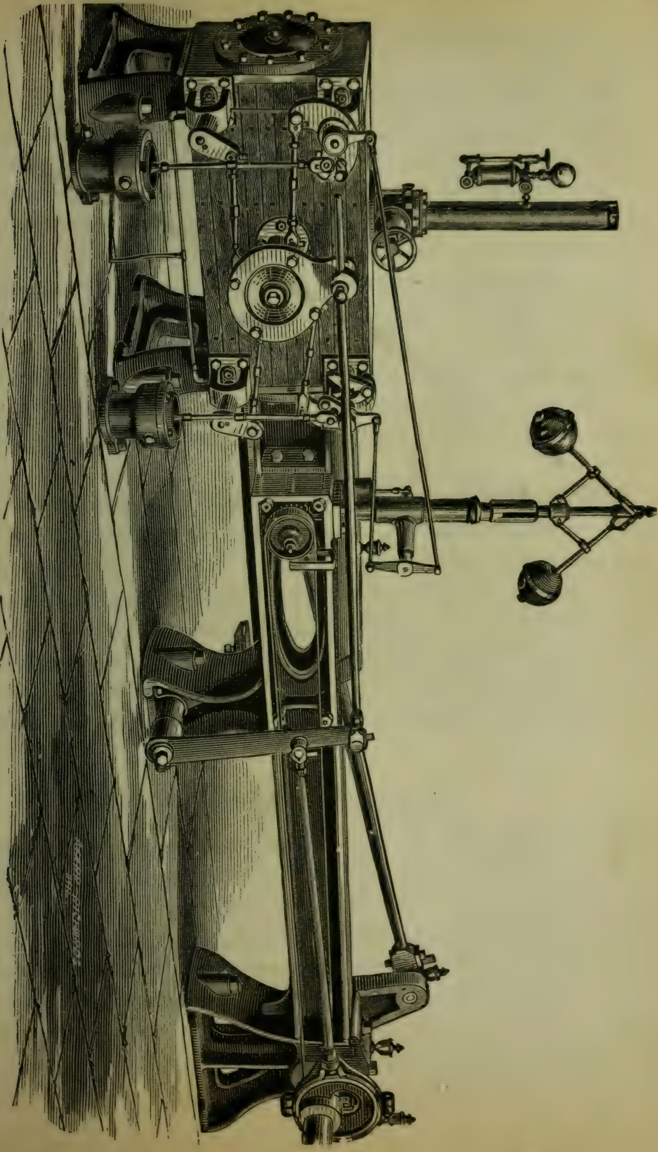
The Reynolds Corliss Engine.

The cuts on pages 178, 179, represent the front and back views of the Reynolds Corliss Engine. It will be observed that the frame is of the girder pattern, the front end of which is faced up to receive the cylinder and slides, while the back end contains the pillow-block bearing; the whole being supported by three pair of legs, which insures rigidity and prevents the possibility of springing, in case the engine should be run at a high rate of speed or loaded beyond its rated capacity. The cross-head has its support on the slides, directly opposite the centre of the cross-head pin, thus avoiding the springing and final breaking of piston-rods, as is often the case where the support is carried back of the centre of the cross-head, as is done in most engines of this type. It is provided with convenient mechanical arrangements for easy and accurate adjustment in case of wear.

The valves are of such construction that they have double the wearing surface ordinarily found in engines of this type. This obviates the rapid wear of the seats, which must occur where the wearing surfaces are small; while, in consequence of the peculiar construction of the valve-gear of these engines, they can be run at any desired speed. The valves open with perfect regularity and close instantaneously, which is a feature of great importance in itself, especially in flouring-mills, as it admits of the line-shafting being coupled directly to the engine-shaft, thus avoiding the use of expensive counter-gearing, and still giving the fly-wheel sufficient motion to properly "lead" the stone and avoid "backlash." No springs are required on either steam- or exhaust-valves. The steel catches used for opening and liberating the steam-valves are so constructed and arranged as to give eight wearing faces on each piece; while by unhooking the eccentric-rod, all the valves can be easily moved and the engine worked by hand, which prevents the liability of its catching on the centre, which is a source of annoyance, especially in the case of large engines. The liberating portion of the valve-gear is claimed to be an improvement on any



Front View of the Reynolds Corliss Engine.



Back View of the Reynolds Corliss Engine.

other arrangement employed on any Corliss engine in use at the present day.

Some of the most important features of the Reynolds Corliss Engine are that they are stronger and heavier than most engines of that class; that the valves are under the complete control of the governor, which is very powerful and sensitive, thus insuring uniformity in speed, which is a feature of great importance for milling and most other manufacturing purposes; that the valve-gear is simple and conveniently arranged for accurate adjustment; that the fly-wheel is turned on the face and sides and accurately balanced; that the wearing surfaces, whether revolving or rubbing, are ample, which prevents the possibility of rapid wear and the expense of repairs; and that the cross-head pin, crank-pin, and piston-rod are made of steel, and the crank-shafts of the best hammered iron.

The Reynolds Corliss engines are in very general use, and have a well-earned reputation for durability, efficiency, and economy. The condenser and air-pump are new in design, simple, and efficient; in fact, the whole design and arrangement of these engines show them to be the result of mature mechanical deliberation. They are manufactured, both condensing and non-condensing, simple and compound, of any size and power, to meet the requirements of purchasers, by Edward P. Allis & Co., Milwaukee, Wis.

Steam- and Exhaust-Pipes.

The diameter of the steam-pipe varies with leading engine builders between $\frac{1}{4}$ and $\frac{1}{3}$ the diameter of the cylinder, the exhaust-pipes being from about 30 to 50 per cent. larger. Some builders make them little, if any, larger; but too small steam- and exhaust-pipes are a prevailing vice amongst small builders, especially those in country districts, who do not use an indicator to determine their proportions. The proper diameter for steam- and exhaust-pipes may be found by *multiplying* the diameter of the piston in inches by its speed in feet per minute, and *dividing* the product

by 1440 for steam- and 1140 for exhaust-pipes; the quotient will be the diameter of the pipes in inches. For short and direct pipes, however, the divisor may be increased to 2000 for steam- and 1440 for exhaust-pipes. These latter divisors will give proportions a trifle larger than the average, especially for exhaust.

Rock-Shafts.

Some engine builders make the diameter of the rock-shaft $\frac{1}{4}$ the diameter of the crank-shaft; if subjected to torsion, it should be $\frac{3}{8}$, and in some cases $\frac{1}{2}$, the diameter. The torsion on a shaft is in proportion to the length of the arm to which the valve is attached. About 10 times the area of the slide-valve in square inches will nearly equal the force in pounds required to move it under 100 pounds steam pressure, though, when dry or starting, it may amount to 12 times or more. The diameter of a rock-shaft may be found by the following rule. *Multiply* the maximum resistance in pounds by the length of the arm which divides the valve, and *divide* the product by 128; the *cube root* of the quotient will be the diameter of the shaft in inches. The size thus found will answer for ordinary wrought-iron shafts, and will resist greater strain than the above rule provides for. The rocker and rock-shaft are being fast superseded by the guide-block.

Cross-Head Bearings.

The area of the wearing surface of a cross-head (that is to say, $\frac{1}{2}$ the total, above and below) should not be less than $\frac{1}{3}$ the area of the piston, nor ever exceed $\frac{1}{2}$ of it. Many steam-engine builders make the length of the cross-head bearings $\frac{2}{3}$ the diameter of the cylinder, and their width $\frac{5}{24}$ of the same, which appears to be a good proportion, and may be *illustrated* as follows: $\frac{2}{3}$ of a 12 in. cylinder is 8 inches in length, and $\frac{5}{24}$ is $2\frac{1}{2}$ inches in width, which gives 20 sq. inches for each shoe, or 40 for both, which is a good proportion; but it should be slightly greater in the case of

short connected engines running at a high speed. The cross-head gibs are generally termed shoes, and the grooves in which they move are called V's.

Valve-Rods.

The diameter of valve-rods varies for moderate sized engines from $\frac{1}{10}$ to $\frac{1}{12}$ the diameter of the cylinder. Their diameter in any case should be proportioned to the size of the valve, whether it is balanced or not. If $\frac{1}{10}$ the area of the valve be considered as a piston of such area, $\frac{1}{6}$ its diameter will bear about the same relation to its maximum strain as piston-rods do; but valve-rods are generally made somewhat larger than such a rule would give, because they are not so well protected against side strains as piston-rods. Probably, since the area of a piston-rod should be from $\frac{1}{36}$ to $\frac{1}{50}$ the area of the piston, according to its length and material (steel may be smallest), a valve-rod should be about from $\frac{1}{300}$ to $\frac{1}{350}$ of the unbalanced area of the valve for high-pressure engines.

The Eccentric.

The eccentric.—An eccentric is substantially a crank, with its pin enlarged in diameter so as to inclose the shaft on which it is placed within its periphery. It gives exactly the same motion that would be obtained from an ordinary crank of equal throw. The eccentric is sometimes called a cam, which is erroneous, as the latter is always used to obtain a motion different from what can be obtained from a crank. The term "cam," when used without qualification, is indefinite, and conveys no impression of its precise form or functions. It is a mechanical element of such a form that a solid body held against, but not revolving with, the periphery of contact may have an intermittent, alternating motion.

Fore eccentric. — A "term" applied to the eccentric, which is connected by its rod to the upper part of the link, to move the valve for the forward motion; but the reason that the forward motion is derived from the upper end of the link arises from

convenience, and not from necessity. The reverse conditions could be introduced very easily.

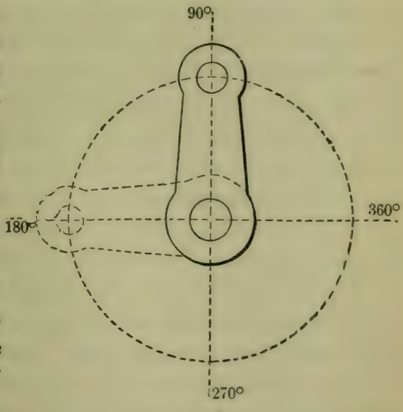
Back eccentric. — The eccentric connected to the lower end of the link by which the valves are adjusted for the backward motion.

Throw of the eccentric.— The "term" throw of the eccentric is understood to be the same as the travel it imparts to the valve, and which is understood to be equal to the width of both steam-ports with the lap added.

Angular advance of the eccentric means the angle at which it stands in advance of that which it would occupy if the valve were in the centre of its travel, and the crank at its centre.

The Crank.

The generally prevalent idea among mechanics that there is an actual loss of power in the use of the crank, has stimulated inventors to substitute for it a device that would utilize all the power exerted against the piston without loss. As a result, the U. S. Patent-Office, as well as those of the different countries of Europe, are crowded with arrangements intended to supersede the crank; the most popular, and consequently the most frequently resorted to, being the rotary engine, in which the effective force of the steam would be constant, while in the case of the crank it is intermittent;



but, so far, no rotary arrangement has ever been able to compete, in point of economy, with the reciprocating motion of the crank.

Strictly speaking, there is no loss of power in the use of the crank, as, while there is a great variation in the power a given

pressure of steam can exert at different points of the stroke, it is known that when the power is least the consumption of steam is least. Suppose an engine has 2-feet stroke, the piston would travel 4 feet for each revolution; during each stroke the effective length of the crank varies from 0 to 1 foot; its average effective length would be equal to the radius of a circle whose circumference was 4 feet, or 7·68 inches. The power of the engine would be the same as if it acted on a constant crank of 7·68 inches, and the displacement, and consequently the consumption of steam, would be the same as before.

If the piston acted on a constant or average crank of 12 inches in length, it must travel a distance equal to the circumference of a 24-inch circle, or $63\frac{3}{8}$ inches. Though such an engine would have proportionately more power at the same number of revolutions, it would consume proportionately more steam. The power of a crank is greatest for early cut-offs at the point at which the valve closes, and for late cut-offs when it stands at right angles with the connecting-rod, which point, as may be seen from the cut on page 175, is not in the middle of the stroke.

An examination of the connecting-rod of an engine in motion, will show that the two ends pass over different spaces in a given time. If, for instance, in one stroke the end of the connecting-rod that is attached to the cross-head moves through one foot, the end which is attached to the crank-pin, and makes a half revolution in the same time, passes through 1·5708 feet. Suppose that an engine is placed with its crank on the centre, and steam is admitted; no motion will be produced, and consequently there will be no power developed, and no expenditure of steam; but let the piston make a stroke, the power exerted is equal to the force or pressure acting on the piston multiplied by the space passed through, or it will be 100 foot-pounds, assuming the data previously given. During the same time the crank-pin has passed through a space of 1·5708 feet, and the force or pressure exerted has been 63·66 pounds, so that the power exerted during this time, or the product of 1.5708 multiplied by 63·66, is 100 foot pounds.

The boss of the crank is that part into which the shaft is inserted, and which butts against the main-bearing. In common practice, its width, when of cast-iron, is about twice the diameter of the crank-shaft journal, and the width at the pin is generally about twice the diameter of the pin. The section of the crank between the shaft and the pin is termed the *web;* its area is generally equal to that of the crank-shaft. When the crank is round, it is called a crank-plate, or disc. The only advantage that the circular possesses over the ordinary form is that it affords better facilities for balancing.

Crank-Pins.

Probably no part of the steam-engine more imperatively requires perfection in material and workmanship than the crank-pin, if cool, noiseless running is considered desirable. Yet it would be safe to say that the cranks of most engines are so imperfectly fitted as to be out of line with their shafts. The most frequent causes of trouble with crank-pins are lack of parallelism between the pin and shaft, imperfect material, untrue turning, and inadequate wearing surface.

It is generally understood that, when a pressure exceeding about 800 lbs. per square inch is imposed upon a journal, lubrication with oil is no longer adequate to prevent destructive wear. In the case of crank-pins this limit is frequently approached, and in some cases exceeded. Very few engine manufacturers make their crank-pins exceed *one-fourth* the bore of the cylinder in diameter and *one-third* of it in length, the majority being short of this proportion.

Assuming this proportion, and that the rule for finding the effective wearing surface of a journal is to multiply its diameter by its length, a little calculation will show that the area of the piston exceeds the wearing surface of the pin over $9\frac{4}{10}$ times. Then suppose the piston to be subjected to a pressure of 85 lbs. per square inch, which is not unusual, the pressure on the crank-pin will be $85 \times 9{\cdot}4 = 799$ lbs. per square inch. If such a pressure was constant, it is very probable that no material, perfection of

workmanship, or lubrication would prevent the heating and speedy destruction of the pin and boxes; but in the case of the crank-pin such pressures are but momentary, and do not last long enough to allow destructive wear to begin. The alternating intervals of no pressure assist in the necessary redistribution of the lubricant; still, when we multiply the mean pressure on the piston by the number of times that its area exceeds that of the pin (ten times, in many cases), the wonder will be not that so many pins give trouble, but that so many do not. An increase in the dimensions of the pin would, it is true, proportionately diminish the pressure per square inch; but the loss of power, by the increased friction thus induced, would be equally as objectionable as the evil which it was intended to remove.

The length of a crank-pin should be equal to the horse-power of the engine divided by the stroke; the quotient multiplied by a coefficient which has been found by experiment to range from 1·3 to 1·5. For instance: if a crank-pin is required for an engine $24'' \times 48''$, capable of developing 250 Hp., then $250 \div 48 \times 1{\cdot}5 = 7{\cdot}81$, or $7\frac{13}{16}$ in., which is the required length.

To determine whether a crank-pin is in line with the centre of the cylinder or not, put on the connecting-rod and key the box up snug on the pin; then disconnect the rod from the wrist of the cross-head and move the crank round, and if the rod maintains a central line in whatever position the crank may be placed, the crank-pin is in line with the centre of the cylinder. This test will also serve to prove the correctness of the boring of the pin-boxes. If they are not bored exactly at right angles to the centre line of the rod, troubles similar to those caused by an untrue pin will ensue. Another oversight not generally thought of, and which causes much trouble with crank-pins, is that, in planing off the stub-ends of the connecting-rod, the machinist, through ignorance or inattention, planes more off one side than the other. As a result, every time the rod changes its position, the box will pinch on the crank-pin, and cause undue heating.

Crank-Shaft Journals and Main-Bearings.

The conditions so essential in the manufacture of crank-pins, viz., good material, excellent workmanship, and accurate fitting, hold good also in the case of crank-shaft journals. Unlike the crank-pin, steel is not used in the case of shafts, principally in consequence of its extra cost; therefore forged or rolled iron is the material most generally employed. Since wrought-iron is never found perfectly homogeneous, the difficulties which lie in the way of a perfectly true and cylindrical journal are much greater than with a steel crank-pin.

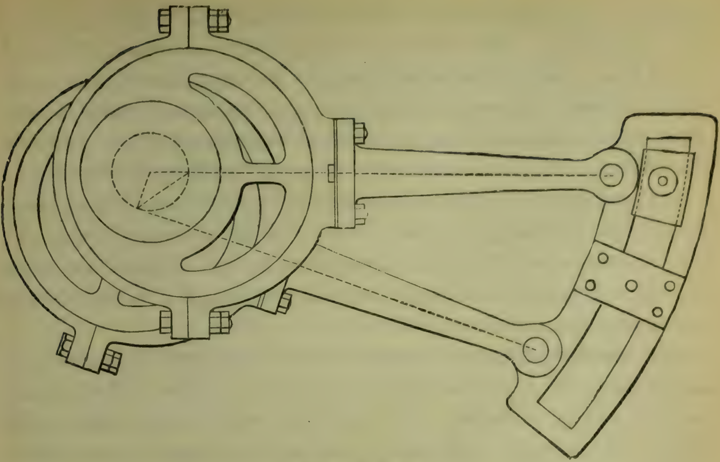
When the initial pressure on the piston is 80 lbs. per sq. inch or upwards, the diameter of the crank-shaft bearing should not be less than ½ the bore of the cylinder; and, in order to prevent springing, it should be as near the centre-line as possible. Although it will be impossible to entirely prevent springing, with high speed and initial pressure, yet, by this arrangement, the liability to spring may be very much diminished. The length of the crank-shaft journal should not be less than twice its diameter, though some engine builders of good repute make them shorter. The longer the journal, within reasonable limits, the more durable it will be, providing the shaft does not spring, and is always perfectly in line with its bearings. But as these conditions cannot be always realized for any length of time, it is not advisable to attempt any greater length than the foregoing.

As the pillow-block bearing is not self-adjusting, it is of great importance that it should be perfectly true with the line, so that the contact of the shaft may be as nearly even as possible throughout. The most general construction consists of a bottom box, side and quarter boxes, adjusted by set-screws, or wedges and a cap; but the simple box and cap, parted at an angle of about 30° from the perpendicular, with its bolts as short as possible consistent with requisite strength, possess the important advantages that they do not tighten on the journal when it begins to heat, as is the case with many of the ordinary forms in use, and that at that angle

the compensation, both horizontal and vertical, may be better provided for. The outer pillow-block bearing being subjected to less severe wear, does not require the same care in its proportions and finish, but should not, for that reason, be slighted.

Keys, Gibs, and Straps.

The key, gib, and strap are the most simple and effective mechanical devices which could be employed for securing the connecting-rods of steam-engines to the wrist- and crank-pins, and taking up the lost motion in the boxes, as they possess sufficient strength without extra weight of material, and facilitate quick and easy adjustment. There is quite a wide difference of opinion among builders in proportioning the keys, gibs, and straps of their engines. Some make the thickness of both straps on the connecting-rod $\frac{1}{2}$ the diameter of the crank-pin, and their width about $\frac{3}{4}$ the length of the pin; while others make the width of their straps three times their thickness, and the area of the cross-section at the mortise equal to the area of the smallest part of the connecting-rod; while others, still, make them equal in strength to the weakest point in the piston-rod, which they undoubtedly should be in any case. It has been customary, heretofore, to make straps thinner at the yoke than at the mortise; but this has been partly abandoned, as the amount of material saved was insignificant, while the extra work was considerable. The depth of the gib and key in a good engine is generally about three times their thickness, and the taper at about 1 in ten; though it ranges all the way from the latter to 1 in 24, 1 in 15 being about the average. It is customary in some instances, as in the case of marine engines, locomotives, and other fast-running engines, to pass a bolt, and in some cases two, through the stub-end and straps, as a precaution against accidents; the holes in the straps being the exact size of the bolt, while those in the stub are slotted, for the purpose of admitting of adjustment by the key and gib.



The Link.

The link-motion is an arrangement of valve-gear for reversing engines and varying the rate of expansion. It consists of two eccentrics, with straps and rods. The eccentrics are so placed that when one is in the right position for the engine to move forward, the other is in the position for moving backward; and by raising or lowering the link, motion will be communicated to the valve and the engine will move backward or forward. The result of this combination is that the link receives a reciprocating motion in its centre; since, when one eccentric is moving the end of the link in one direction, the other is moving the other end in the other direction; so that the link will have nearly the same motion communicated to it as if it were suspended from a pivot at its centre.

The horizontal motion communicated to the link by the joint action of the eccentrics, is a minimum at the centre of its length, where it is equal to twice the linear advance, and it increases towards the extremities of the various periods of the block in the link, or of the link on the block, on the general principle that admission varies with the travel of the valve. The nature of the motion derived from the link is modified by the positions of the

working centres, and most especially of the centres of suspension and connection. The centre of suspension is the most influential of all in regulating the admission, and its transition horizontally is much more efficacious than a vertical change of place to the same extent, inasmuch as the vertical movement of the body of the link, with the consequent slip between the link and the block, is the least possible when the suspended centre lies in the centre line of the link, and increases as the centre is moved laterally. The centre line of the link is therefore, in this respect, the most favorable location for the suspension, even though it be not always practicable for equal admissions.

The amount of travel communicated to the valve depends upon the distance the block is from the centre of the link. By moving the link up or down on the block, the travel of the valve will either be increased or decreased; and since the travel of the valve is the measure of the lap, to reduce the travel is tantamount to increasing the lap, and also the lead. Thus the link-motion becomes an expedient for regulating the amount of expansion with which the engine works. Though it may be claimed by some that cutting off by the link has a tendency to affect the exhaust, it does not do so to any injurious extent, as the later opening of the exhaust is a positive advantage, as it balances the resistance due to the early admission of the steam at the other end, before the engine has reached the end of the stroke. It will be seen, for the foregoing reasons, that the link is a perfect expansion-gear, as, when in full stroke, it is superior, in many respects, to most other cut-off devices, since, while the lead is increased as the travel of the valve is decreased, or, in other words, as the link is lifted towards the centre, and the supply of steam cut off at an earlier point in the stroke, the lead becomes a positive advantage, as it serves as a cushion to the piston when its reciprocating motion is rapid, as is frequently the case.

The ease and facility with which the link may be handled is another very important feature in its favor. In fact, what could we do without it when handling engines, especially large locomo-

tives or marine engines, which have of necessity to run backwards with the same ease, speed, and facility as they run ahead? The link is a splendid mechanical conception, and one of the greatest improvements that has ever been made in the locomotive, marine engine, or any other class of motors requiring a reversing gear.

The radius of the link is the distance from the centre of the driving-axle, or shaft on which the eccentric is located, to the centre of the link; while the link itself is a segment of the circle of that diameter. The length may be longer or shorter; but any variation from these proportions will give more lead at one end than at the other while working steam expansively; but the radius may be several inches shorter or longer, without materially affecting the motion. The vital point in designing a valve link-motion is the point of suspension of the link. If it is suspended from the centre, it will invariably cut off steam sooner in the front stroke than in the back stroke, while working expansively.

The nearer the block is brought to either end of the link, the greater will be the travel of the valve, and the more the steam and exhaust will be opened. The *term* "full-gear forward" means that the link is dropped to its full extent; while "full-gear backward" means that the link is lifted to its full extent. When the link-block stands directly under the saddle-plate, both ports are closed, and neither admission nor exhaust can take place. The distance between the block and the end of the link when in full-gear is termed the clearance.

In the Walschäert link-motion, which was used on one or two of the small engines at the Centennial Exposition, the mid-gear movement was derived directly from the cross-head, while the end, or full-gear, movement was derived from a single eccentric, or a return crank, from the main crank-pin. The middle of the link is stationary, and, of itself, imparts no motion to the valve; but between the link and valve is an arrangement for imparting a reduced and reversed copy of the piston movement to the valve, which movement, being always present, modifies that of the ec-

centric at all points, giving it the effect of angular advance, which is not given to the eccentric in the case of the ordinary link-motion.

Lifting and stationary links. — The lifting-link is raised and lowered to effect the changes it is designed to perform; while in the stationary link the block, instead of the link, is shifted. In the stationary link but one eccentric is generally used, the throw of which corresponds to the middle of the ordinary link; for this reason, more mischief would be caused by any lost motion in the eccentric straps or other connections. Moreover, it does not allow of ready, independent adjustment of the backward and forward motion in full gear.

Fly-Wheels.

The object of the fly-wheel is to equalize the motion whenever either the power communicated or the resistance to be overcome is variable. In the one case, the fly-wheel may be said to be a distributor of power. The complicated impulses, acting on the mass in motion, preserve the momenta, without disturbing the regularity of movement. The effect of one impulse is so absorbed or distributed in the momentum of the wheel, that it may be said to have hardly been diminished before the next impulse is received.

In the other case, or where the fly-wheel is used to overcome a variable resistance, it may be considered a conservator of power. The power having been exerted in getting up the speed, is retained in the moving mass, and the whole of the power expended, with the exception of that which has been lost through friction and resistance of the air, can be brought to bear at any instant upon the resistance to be overcome. When the crank and connecting-rod are in one straight line, as they must be twice in each revolution, the crank is said to be on its dead-centre, because there the force of the piston is dead or ineffective. It is evident that, when the crank is at right angles to the connecting-rod, the latter is exerting the maximum of power; but when the forward or back-

ward dead-centre is reached, the crank would remain there, but for the action of the fly-wheel, which, by its accumulated momentum, carries it over the dead-centre.

Thus, through the momentum of the fly-wheel, no perceptible variation occurs in the velocity of the engine; the unequal leverage of the connecting-rod is corrected, and a steady and uniform motion produced. The fly-wheel, as before stated, is a regulator and reservoir, and not a creator of motion. The accumulated velocity in the fly-wheel, where the motion is required to be excessively equable, should be about six times that of the engine when the crank is horizontal. As regularity of motion is of much greater importance in some cases than in others, the weight and diameter of the fly-wheel must depend on the work and the character of the machinery it is intended to drive; so that, in proportioning a fly-wheel to a given engine, attention must be paid to many particular circumstances rather than to any given rule. There are circumstances in which the use of a fly-wheel may be dispensed with, as where a pair of engines work side by side, whose cranks are at different angles, so that one assists the other to pass the centres, or where smoothness of motion is not an absolute necessity.

Rule for finding the proper weight of the fly-wheels of steam-engines.

Divide the constant number 7,000,000 by the square of the number of revolutions per minute, and by the diameter of the wheel in feet. The quotient will be the number of pounds per horse-power required in the rim of the wheel.

The above rule is correct, so far as it recognizes the fact that the efficacy of a fly-wheel increases with the square of its velocity and with its diameter. The constant number is found by taking some engine whose fly-wheel is known to be right at a given load, dividing its weight by the horse-power developed, and multiplying the quotient by the square of the number of revolutions per minute, and by the diameter. When so found, it will give correct results for all other engines of the same class doing similar work.

This constant number must not, however, be regarded as arbitrarily fixed. It will give the weight of the wheels near enough for automatic cut-off engines.

The Watertown Automatic Cut-Off Engine.

The cut on the opposite page represents the Hampson Automatic Cut-Off Engine, the bed-plate of which, as will be observed, is of the box pattern; the metal in which is so distributed as to combine strength, stiffness, and rigidity, without extra weight. The steam-cylinder and main pillow-block and guides are attached to the bed-plate in such a manner as to prevent the possibility of becoming loose when the engine gets out of line. As the steam-chest is the full length of the cylinder, with the ports opening directly from it into the clearance, it enhances the value of these engines very much, as it obviates the waste induced by long steam-ports.

The valve-gear receives its motion from two eccentrics on the main shaft, the one next the pillow-block being connected with the main valve, which is an ordinary slide-valve, with this exception — that the steam, instead of passing in at the ends of it,

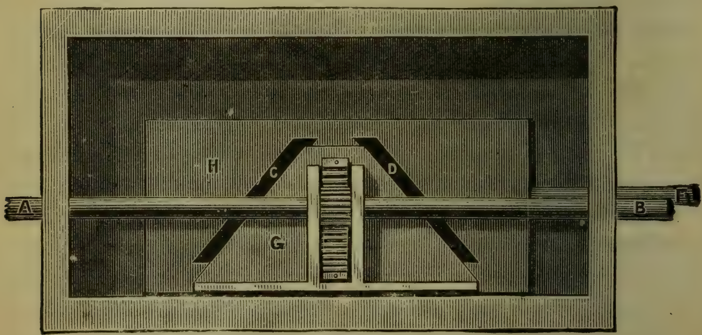
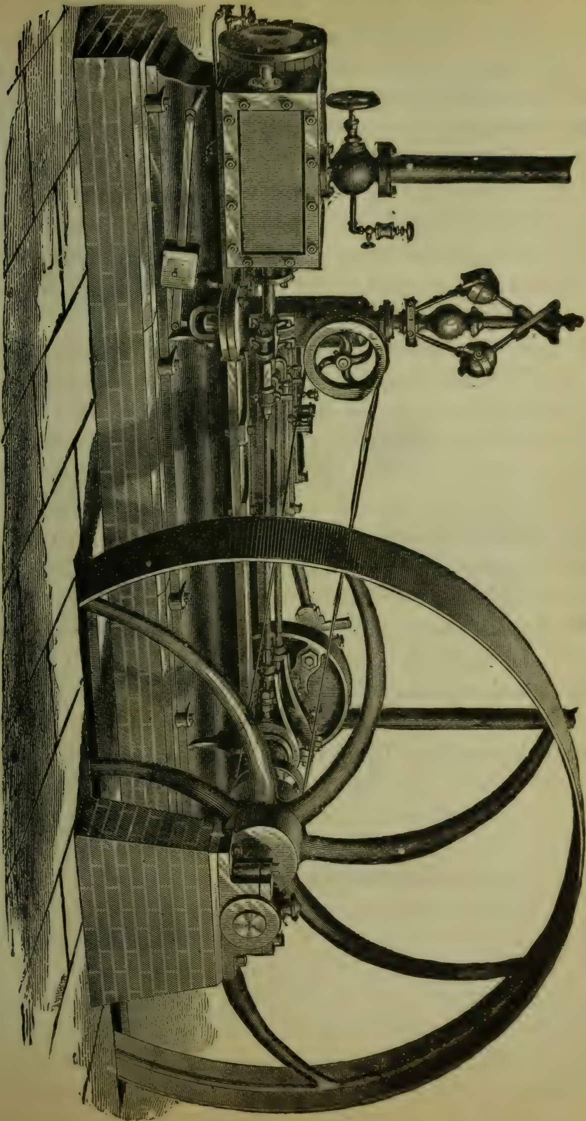


Fig. 1.

enters through it by means of ports, as shown at *C D*, Fig. 1. *H* represents the back of the main valve, which is also the seat



The Watertown Automatic Cut-Off Engine.

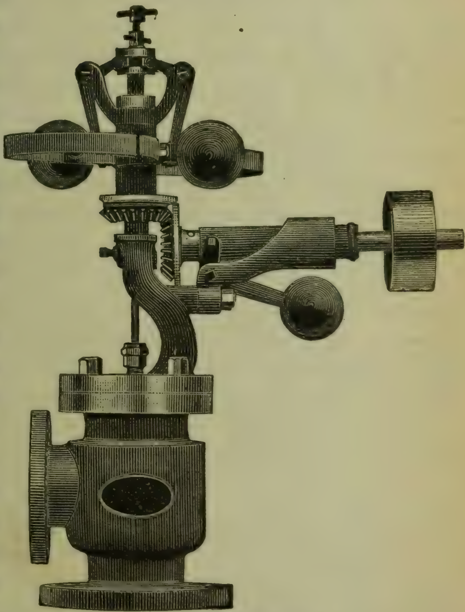
of the cut-off valve, *G*. *F* represents the stem of the main valve, and *B* the stem of the cut-off valve, which is continued on through the end of the steam-chest, and is held steady when the engine is working by means of a horn at *A*. The reader will notice that there is a rack cut in the back of the cut-off, which engages the teeth of a small wheel on the valve-stem, and from this device any one would soon come to the conclusion that the adjustment of the cut-off is accomplished by rolling the valve-stem. This, as a matter of course, will raise and lower the cut-off valve by means of the rack and pinion, thereby opening the ports.

The governor, which is very powerful and sensitive, embodies some peculiarities of design and construction not common in governors, inasmuch as the point of suspension, instead of being on the same side of the spindle as the ball, is carried over to the opposite side, thereby greatly increasing its power and sensitiveness. Directly under the governor there is a disc on the valve-stem, with teeth cut on the periphery about half the circumference, and these teeth engage a rack connected with the governor-spindle. Consequently, as the balls of the governor rise and fall, a proportional movement will be transmitted to the cut-off valve. To determine the point at which the engine is cutting off when running, the plain part of the disc, which is connected with the governor and valve-stem, has marks and figures upon it, each mark indicating a point in the length of the stroke. There is a point which coincides with these marks, and can be seen under the pulley attached to the governor. To increase or diminish the speed, a counterweight is attached to the end of the governor-spindle, under the steam-chest.

These engines possess many excellent features. The bearings are well proportioned and all the parts thoroughly fitted; the fly-wheels are turned on the face and sides and accurately balanced; the connecting-rod and crank-shafts are made of the best hammered wrought-iron; the crank- and wrist-pins are made of steel; the connecting-rod boxes of gun-metal, and the main-bearings lined with the best anti-friction metal; while the cylinder is cast of car-wheel iron, and jacketed to prevent radiation.

Steam-Engine Governors.

The subject of regulating the speed of steam-engines, and more especially those which, from circumstances and the nature of the work to be performed, are liable to constant change, has of late years received no little attention from engineers and practical inventors, and as a result various kinds of governors have been introduced. It would be safe to say that this device has absorbed more thought, and received more attention on the part of mechanics, than any other adjunct of the steam-engine. In the ordinary governor, the principal part of the apparatus consists of a pair of balls revolving round a vertical axis or spindle driven by a train of mechanism, generally mitre-gears, which causes their angular velocity of revolution to bear a fixed ratio to the velocity of the prime mover. The rods of the pendulums place themselves at an angle with the vertical axis, so that the common height of the pendulums is that corresponding to the number of turns in a second. The regulator must be so adjusted as to be in the proper position for supplying the proper amount of power when the pendulum-rods are at the angle of inclination corresponding to the proper speed of the machine.



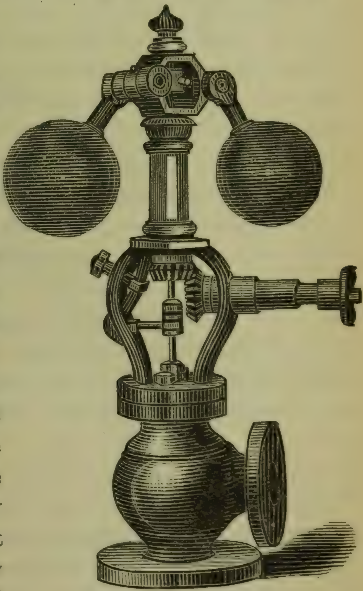
The Waters Governor.

When the speed deviates above or below that amount, the outward or inward motion of the pendulum-rods acts on the spindle, so as to open the valve when the speed is too low, and close it when it is too high.

In the attainment of this object, the principle of centrifugal force, as embodied in the old fly-ball governor of Watt, has been more resorted to than any other; but, aside from this, the governor has been so improved, altered, and reconstructed, since his time, as to be almost unrecognizable; but still the old principle is there, and also the three prominent defects which so materially interfere with its efficiency. The first of these is friction which arises from the joints, and is caused by swinging the balls or weights by the short end of the arm or lever to which they are attached. The second defect is due to the fact that the balls, as they assume different positions in keeping with the speed with which they revolve, are obliged to rise or fall. This is necessary in order that the resistance which the weights offer to centrifugal force should constantly increase; if it did not so increase, the weights, when once started from their position of rest, would instantly go to the extreme limit of motion. The rising of the balls shortens the distance which they are allowed to move for a given variation by bringing the centres of ball and arm on which they swing into a straight line, so that a variation which moves the balls a given distance upward, if it occurs again, will not move them nearly so far in the same direction. Again, the same force that would support the balls in any plane would not raise them to that plane from a lower one. So between friction, which destroys the delicate power that the balls assume under a slight change, and the necessity for a large change to overcome their inertia, it is almost impossible to attain a degree of regulation which would be equal to all requirements.

Governors when attached to throttle-valves work under circumstances that necessitate the use of openings for the passage of the steam that are too small in area, so much so that the useful effects of the steam are considerably diminished. On this depends

the ill repute of throttling engines as compared with those which regulate by governor controlled valve motions or variable cut-off. If the valve of a governor has too large openings, it will, owing to the unsteady action of the governor, admit too large a quantity of steam, and cause a jumping of the engine; then, in trying to shut off this extra amount, it shuts it all off; in fact, the governor cannot fix it exactly right, being incapable of delicate changes. This difficulty is best met by making the openings in the valve of peculiar shape, so that they open and close in a ratio different from that of the governor. With a governor that would run perfectly up to theory, and be steady and capable of taking a position in keeping with the speed, and not leaving it without a change in speed, a very large area might be used, and the useful effects of the steam would not be impaired, neither would there exist a necessity for great changes in speed to get the required opening and closing of the valve. The extra amount of steam required to drive a heavy addition of load on an engine is surprisingly small, provided that the engine can get the steam at the very instant the load is applied, and before the momentum of the machinery becomes much reduced; but let the engine once get below speed, the circumstances will be very different, as, even without any load, the engine would take some time to come to speed.



The Shive Governor.

The third defect in governors on throttling engines is that the spindle or valve-stem has of necessity to pass through steam-tight,

packing- or stuffing-boxes, which have to be screwed up to prevent leakage, without any guide save the judgment of the engineer, which increases the friction and interferes with the free action of the governor. There is also the friction on the governor-valve necessary to overcome the power required to move the valve-stem through all its bearings, stuffing-boxes, guides, etc., under the pressure of steam. Were it possible to construct a governor for throttling engines which would approach in practice what theory would demonstrate, the fly-ball or centrifugal governor would be a perfect regulator; but this appears, according to mechanical laws, to be impossible. By the use of isochronous governors, which would not admit of any variation of speed, but would be in equilibrium at any speed, whether the balls were up or down, or in any other position, the defects of the common governor were supposed to be obviated; but it was found by experience that power and stability were necessary, and isochronism in its strict sense unattainable.

The economy of a good governor has rarely been appreciated by owners of steam-engines and steam-users. Experience has shown the speed best adapted for each and every process in the manufacturing and mechanical arts, and the governor that fails to meet all the varied requirements of each process is of no value in an economical point of view. Every stroke which an engine makes below its regular speed increases the cost of production, and every stroke above it is a waste of steam, and consequently of fuel. If an engine is geared to run at 80 revolutions per minute, when a heavy piece of machinery is thrown off, the governor admits of an increase of speed of from 10 to 15 revolutions per minute. This incurs a waste of power, and consequently a waste of from 12 to 20 per cent. of fuel. On the other hand, when a heavy piece of machinery is thrown on, the governor allows the engine to lag behind its regular speed by from 10 to 15 strokes per minute; this increases the cost of production. If a governor is unreliable, it is worthless; if reliable, its first cost is merely a nominal consideration. There are many processes, such as mill-

ing, weaving delicate fabrics, printing from small type, or the very accurate turning of fine material, where a good governor is of immense value. Unfortunately for the progress of the mechanical arts, no governor yet invented has met all the necessary requirements, or the varied circumstances under which they are employed.

Governors are sometimes attached to marine engines for the purpose of equalizing the revolutions in heavy sea-ways, and preventing the engines from racing, which is caused by an insufficient immersion of the paddle-wheels or propellers, and which may be ascribed either to the lightness of the load or the heavy swell of the sea. But from whatever cause racing may occur, it is always attended with danger, as the undue strain to which the machinery is subjected is liable to result in a breakdown. Marine governors have not proved a success up to the present time, nor has any one yet been invented which may be adapted to all classes of marine engines.

Governors should be kept perfectly clean and free from accumulations induced by the use of inferior oil, as such gummy substances have a tendency to interfere with the easy movement of the different parts. Many first-class regulators have been condemned as not being capable of controlling the engine at a uniform speed, when all that was required was a good cleaning.

Governor-spindles working through stuffing-boxes should be frequently and carefully packed, as, when the packing becomes old and dry, if screwed up to prevent leakage, it interferes with the free action of the governor.

Rules for calculating the size of pulleys for governors.—*To find the diameter of the governor shaft-pulley.* Multiply the number of the revolutions of the engine by the diameter of the engine shaft-pulley, and divide the product by the number of revolutions of the governor.

To find the diameter of the engine shaft-pulley.—Multiply the number of revolutions of the governor by the diameter of the governor shaft-pulley, and divide the product by the number of revolutions of the engine.

How to Balance the Reciprocating and Revolving Parts of Vertical Engines.

If the counterweight be so arranged as to describe a circle of the same radius as that of the crank-pin, it must be as heavy as the piston, connecting-rod, and crank-pin; but if it has a greater circle than that of the crank-pin, it may weigh less than the piston and its connections; the only material condition being, that the momentum, or amount of mechanical power resident in the counterweight when moving in one direction, shall balance the momentum of the piston and its connections when moving in the opposite direction.

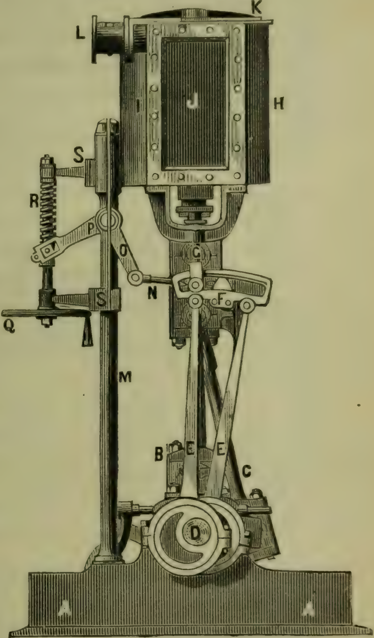
When a vertical engine runs slow, the weight of the piston and piston-rod, cross-head, connecting-rod, and crank-pin must be counterbalanced so that it will stand still in any position; but when the speed is very high, it is necessary to counterbalance only such parts as revolve round the centre of the shaft, the crank-pin, the stub-end, and half the connecting-rod. It is customary to give more steam-lead on the valve at the bottom than at the top of the cylinder in order to compensate for the weight of the piston.

Heating in Journals and Reciprocating Parts of Steam-Engines.

Heating in the journals and reciprocating parts of steam-engines may be attributed to the following causes: bad proportion, improper fitting, unsuitable material, want of homogeneity between the materials of which the journals and bearings are composed, the reciprocating or revolving parts being out of line, the boxes being screwed down or keyed up too tight, dirt, sand, or grit getting into the journals, want of proper lubrication, etc. The last mentioned cause is much more complicated than would at first sight appear, as there are many conditions to be taken into consideration, among which may be enumerated weight of load, area of surface subjected to pressure, velocity of movement, etc.

Reversing-Gear for Marine Engines.

In the early days of the steam-engine, the only reversing-gear in use was the V hook, which was very imperfect, uncertain, and unreliable in action and difficult of adjustment, and, in consequence of its action being positive, steam could not be worked expansively on engines on which it was employed. A more modern arrangement was the loose eccentric, which, in consequence of the eccentric-hook being thrown out of gear, moved half-way round on the shaft whenever it became necessary to reverse the engine. The most perfect reversing and expansion gear ever employed in connection with the steam-engine is the link.



A A shows the bed-plate; *B*, the pillow-block bearing; *D*, the shaft; *E E*, the eccentric-rods; *C*, the connecting-rod; *F*, the link; *G*, the cross-head wrist; *J*, the bonnet of steam-chest; *I H*, the steam-cylinder; *K*, the cylinder-head; *L*, the steam-pipe; *M*, the front column which supports the cylinder; *N*, the reach-rod; *O*, the reverse arm of the bell-crank by which the link is moved back and forth; *P*, the lifting arm; *R*, the screw by means of which the link is reversed; *S S*, the guides through which the spindle of the screw, *R*, moves; *Q*, the hand-wheel by which the lifting arm, *P*, is moved up or down for the purpose of changing the position of the link.

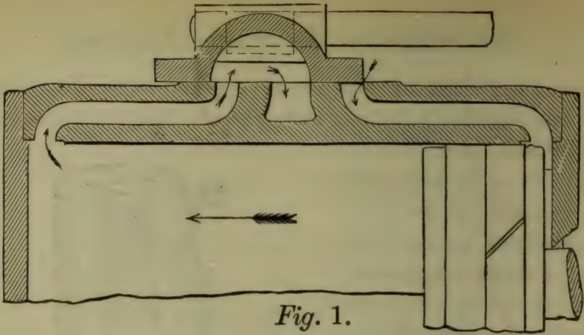


Fig. 1.

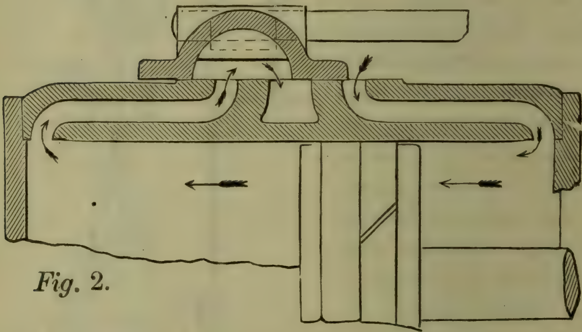


Fig. 2.

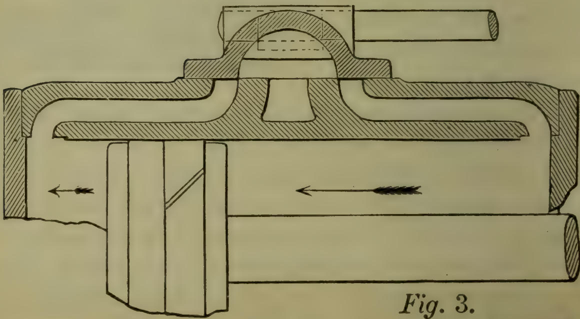


Fig. 3.

The annexed Cuts show the Position of the Slide-Valve at Three different Points in its Travel. (*See explanation on page* 206.)

The Slide-Valve.

The function of the common slide-valve is to admit steam to the piston at such times when its force can be usefully expended in propelling it, and to release it when its pressure in the cylinder is no longer required. Notwithstanding its extreme simplicity as a piece of mechanism, no part of the engine is more puzzling to the average engineer when the problem to be solved is to determine beforehand the results which will be produced by a given construction and adjustment, or the proportions and adjustment required to produce given results. All who have had any experience in constructing and setting slide-valves are aware, in a general way, that the events of the stroke cannot be independently adjusted; that, for instance, a cut-off earlier than about three-fourths of the stroke can only be had at the expense of more or less distortion of the other events, and that for some reason, not always apparent, it is impossible to completely equalize the events of the two strokes, occurring during one revolution.

But hitherto no simple means have been given by which to determine exactly the degree in which a given change in any event affects the rest. There is no lack of literature on the subject, but the manner in which it is generally treated is calculated to bewilder the average reader more than to assist him; to invest the subject with additional difficulties rather than to simplify it. The manner in which the slide-valve performs its functions cannot be at once perfectly shown without the aid of a working model, but a considerable step may be taken in this direction by the construction and study of diagrams similar to the following. It should be understood, however, that the measurements given of lead, cut-off, compression, etc., are only approximately correct; the object being to give the methods by which correct results may be obtained rather than the results themselves.

Fig. 1, page 207, represents the position of the piston and valves at the beginning of the stroke, when the latter is just commencing to open. The motion of both, as will be observed, is to the left.

Fig. 2 shows the relative position of the piston and valve at about $\frac{1}{4}$ of the stroke; supposing the travel to be equal to the sum of the width of both steam-ports and the steam-lap at both ends, so that the ports will be just opened full for the steam, the valve will be moving to the left. When the piston reaches the left end of its stroke, the valve will have moved to the right till it begins to admit steam, at the left hand end, just as Fig. 1 shows the admission taking place at the other end, and during the return stroke, the conditions represented by Figs. 2 and 3 will follow in succession, at nearly corresponding points in the travel of the pistons. *If the piston* was connected to the crank by means of a slotted yoke, the events of the two strokes would occur at exactly corresponding points in the travel of the piston, but the connecting-rod unavoidably introduces a certain amount of distortion, the nature and extent of which will be explained hereafter. Fig. 3 shows the position of the valve at mid-travel, or when $\frac{13}{16}$ of the stroke is complete. The compression at the left end towards which the piston is moving has just commenced, and the exhaust is about to take place from the other end. The events which occur in connection with the slide-valve, viz., admission, suppression, release, and compression, may be explained as follows:—To find the cut-off, exhaust, exhaust-closure, port-openings, and angular advance which will be produced by a given lap, lead, and valve-travel, the lead and laps being equal at the two ends. Suppose the data to be as follows: valve-travel $2\frac{3}{4}$ in., lap $\frac{6}{10}$, lead $\frac{1}{10}$ in, stroke of engine 24 inches.

Draw the circle *A F B G*, etc., the diameter of which may, when the engine is of considerable size, be equal to the travel of the valve, as in the present case. Draw the line *A B* through the centre of the circle, continuing it beyond *B* to a distance nearly equal to three times the distance *A B*. With the given lap in the compasses, draw short arcs of circles at *D* and *E* from the centre, *C*. Draw lines *a b* and *c d*, parallel to each other, touching the arcs *D* and *E*, equidistant from the intersections of the line *A B*, with the circle equal to the given lead. Set the

compasses to a distance which will be to *A B* as the connecting-rod is to the stroke of the engine, which in the present case is about 7⅛ in.; and with the foot in the continuation of the line *A B*, draw arcs *b e* and *d f*, which will locate the points of cut-off, *f* and *e* on line *A B*, which represents the stroke of the engine as well as the travel of the valve. By constructing and applying a scale, such as Fig. 1, in which the travel of the valve is divided

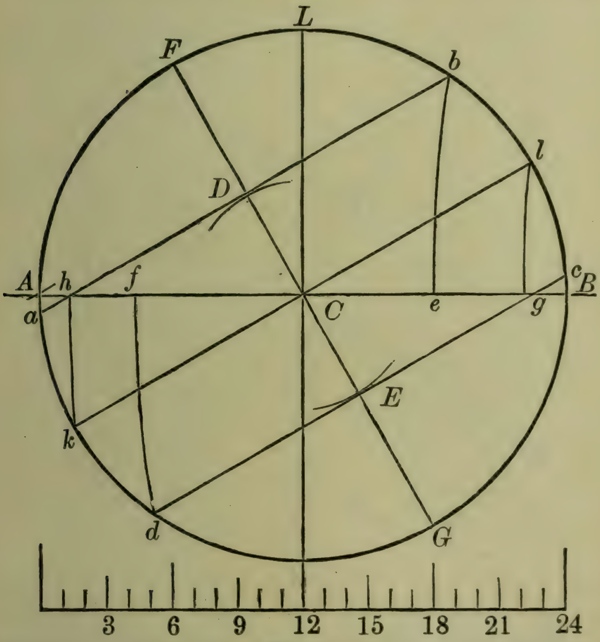


Fig. 1.

into as many parts as there are inches in the stroke of the piston, it is found that, as the piston moves from the shaft, the cut-off, *e*, takes place at 18⅛ inches from *A*, and the other, *f*, at 19⅝ inches from *B*, making an inequality of 1½ inches.

If the valve has no exhaust-lap at either end, working "line or

line," as it is sometimes called, draw line *k l* through the centre *C*, and parallel to *a b* and *c d*, and from *k* and *l* draw arcs *k h* and *l g*, as directed for the arcs *d f* and *b e*, which will locate the points of exhaust and exhaust-closure at *h* and *g*, about $1\frac{1}{2}$ and $1\frac{7}{8}$ inches respectively from the ends of the stroke. *A* represents the end of the stroke nearest the crank; and it will be observed that the events occurring nearest that end are later in the stroke

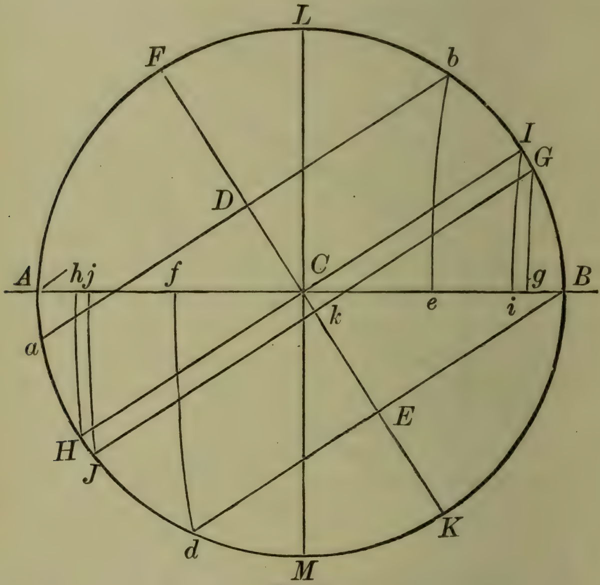


Fig. 2.

than corresponding events at the other end, which will always be the case when the laps are equal at the two ends of the valve. The port-openings will be *D F* and *E G*; *F L* will be the angular advance.

To equalize the cut-off.—By inspection of Fig. 1, it is evident that if line *a b* be moved towards the centre, *C*, the arc *b e* will approach *B*, and cut-off *e*, which is earliest, will be made later

In like manner if line *c d* be moved farther from the centre, cut-off *f* will be made earlier. These changes represent increased lead at *A* and diminished lead at *B*. In constructing a diagram, however, representing a given equalized cut-off, it will be preferable to begin by locating the points of cut-off at the desired part of the stroke, say three-fourths, as at *f e*, Fig. 2, from which points draw the arcs *e b* and *f d*. Then, as inspection of Fig. 1 has shown

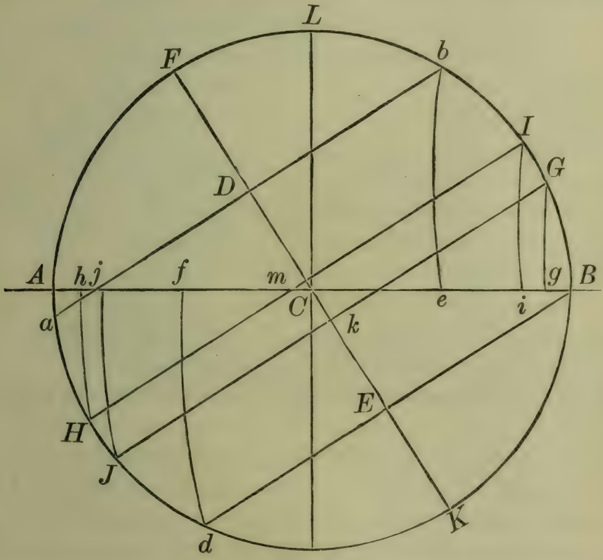


Fig. 3.

that the lead at *B* required diminishing, the lead at that end may be rejected altogether, which will be represented by drawing a line from *d* to *B*. Then a line from *b* parallel to it will cut the circle at *a*, indicating that over $\frac{3}{16}$ of an inch lead will be required at that end (nearest the crank as before explained), against none at the other to equalize the cut-off at three-fourths stroke. From this it will be seen that the cut-off can only be equalized at the expense of the equality of the lead.

To equalize the exhaust at a given part of the stroke. Suppose the desired point be $1\frac{1}{2}$ inches from the end; set off the points at h g, Fig. 2, and draw arcs, h H and g G, as before directed. Then supposing the angular advance, F L, and with it the lines, a b and B d, to have been fixed, draw H I and G J parallel to a b and B d, and from points I and J draw arcs I i and J j, which will locate the points of exhaust-closure at j and i. The distance, C K, will be the exhaust-laps required at the end of the valve next the crank, and the distance of line, I H, from centre, C, will be the negative lap (*i. e.*, the amount less than no lap) required at the other end. To determine whether the distance of a line indicating the exhaust-lap (as I H) from the centre indicates positive or negative lap, observe the effect of increasing its distance from the centre. If the exhaust located by it at one end should be made earlier, and the exhaust-closure located by it at the other end made later, the lap indicated by it is negative. Thus, to move I H farther from C would make exhaust h earlier and exhaust-closure i later; hence it indicates negative lap. The reverse effect would follow by moving G J farther from the centre; hence C K is positive lap.

Fig. 3. To compromise between unequal lead and cut-off.—The lead inequality shown to be necessary, in order to obtain equal cut-off, may be in some cases so undesirable as to render only a partial equalization of the cut-off preferable. Fig. 3 shows such a compromise. It shows that by giving $\frac{1}{8}$ inch lead at A, and none at B, the cut-off will be sufficiently equalized for all practical purposes, as the difference is reduced about one-half as compared with Fig. 1. It will also be noticed that in Fig. 3 the exhaust-lap has been increased to $\frac{3}{16}$ at C K, and about $\frac{1}{32}$ inch at C m, both positive, which gives equalized exhaust-closure at J j, and very nearly equal exhaust at h g. The excess of lead at A over B of course diminishes the lap C D, and increases the port opening, F D, at that end.

Fig. 4 shows the data obtained from Fig. 3 applied to the construction of a common slide-valve. The scale of the valve is

made one-half size for convenience. The valve is shown at mid-travel; *C k* shows the exhaust-lap obtained from Fig. 3 at *C k* and *b m*; that at *C m*, steam-lap; *a E* is obtained from Fig. 3 at *C E;* and *c D* in like manner from *C D*. It will be seen that, notwithstanding there is less steam-lap at *D* than at *E*, the lap *k D* is slightly greater than *E m*, which is due to the fact that the exhaust-lap added at *k*, to equalize the exhaust and compression.

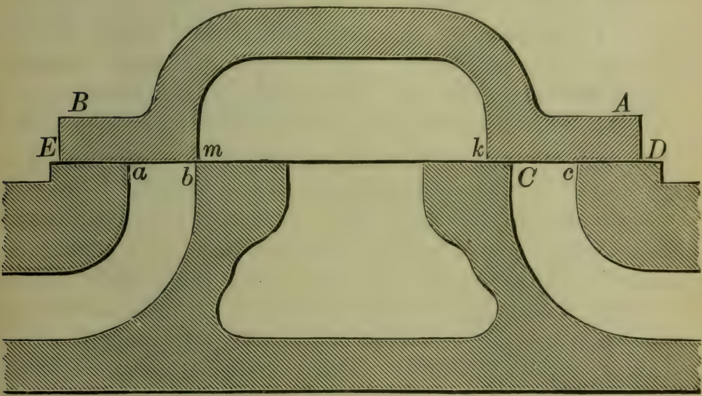


Fig. 4.

slightly more than compensates for the lesser steam-lap at D. If the steam-lap at *D* had been lessened until *k D* equalled *E m*, the cut-off would have been more nearly equal than is shown on Fig. 3, but still not entirely so. From this it will appear that valves may be constructed (as they mostly are) with the two laps equal in width, and in setting them, the exhaust and compression may be equalized, letting the cut-off equalization take care of itself, which it will do by becoming a trifle more than half equalized, as compared with Fig. 1. Such a valve, considered apart from the seat on which it works, would appear to have equal laps of both kinds, and might be so set, as is the case in the adjustment represented by Fig. 1; but, when set to equalize the compression and exhaust, it must be considered as having unequal laps of both

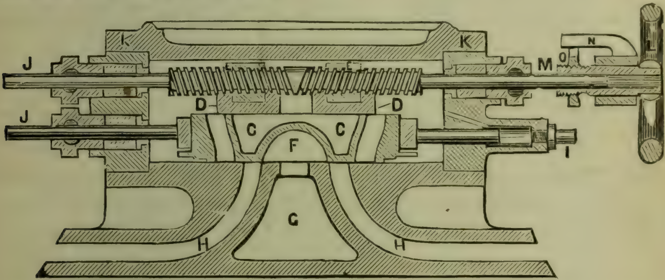
kinds. A valve constructed from the dimensions furnished by Fig. 2, in which, as we have seen, the cut-off, exhaust, and compression were entirely equalized at the expense of lead, equality would have the lap $k\ D$ the shortest.

In determining the best point for exhaust-closure, it should be borne in mind that this event is the one which stands most in the way of an early cut-off, and that it is desirable to know how early it may be located, without detriment to the performance of the engine. The decision of this point will depend mainly on the amount of clearance present. If the clearance is great, considerable compression is not only admissible, but desirable; as the greater the clearance the less the loss of mean effective pressure by the counter-pressure resulting from early exhaust-closure. The steam shut in by the closure of the exhaust is saved, to be used over again during the next stroke; and, when the clearance is great, the loss of power by early compression is more than compensated by the saving of steam; the result is a certain amount of net gain in economy. As a general rule, the maximum compression pressure should not exceed the pressure present in the steam-chest. If it should, the valve is liable to be forced from its seat; and not only so, but the limits within which compression improves the economy would be exceeded.

When the clearance is known, and is reduced to a certain percentage of the stroke, the compression may be fixed at three to five times the clearance, which would, theoretically, raise the compression pressure to from three to five atmospheres; but, in practice, the theoretical maximum is seldom reached. Thus, suppose the clearance of an engine to be equal to its displacement during one inch of its stroke, and the valve to close the exhaust four inches from the end of the stroke; or, in other words, suppose the compression to be four times the clearance, the maximum compression pressure should, theoretically, reach 55 to 50 lbs.; but it will seldom, in practice, exceed 50 lbs., unless the cylinder is jacketed with live-steam, and the valve and piston are very tight.

The proper point to release the steam will depend upon the

travel of the valve, the capacity of the cylinder-ports, and the exhaust-passage in the valve. If these are ample, the release may occur later than when they are not. The point to be aimed at in locating it, is to release in time to avoid any considerable back pressure at the beginning of the return stroke. No responsible engine-builder of the present day will fix on a valve construction and adjustment permanently, until he has first tested its results with the indicator, and satisfied himself that they are the best possible with the slide-valve.



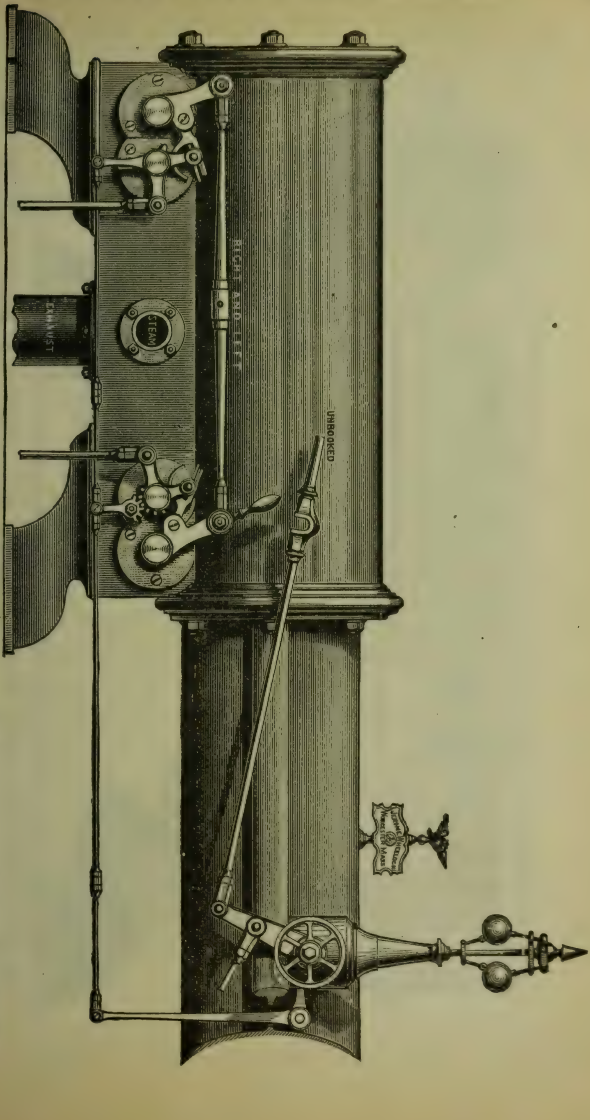
The above cut represents the Myers slide-valve. *C C* shows the main valve, which is whole stroke; *D D* shows the cut-off, what is termed the riding cut-off, because it rides on the back of the main valve, and, as will be observed, the amount of expansion is regulated by right and left hand screws passing through the cut-off valves, and shown above, *D D*. By turning the hand-wheel, *L*, to the right, the cut-off will be decreased, while by turning to the left it will be increased. *H H* shows the steam-ports; *G*, the exhaust cavity, and *F*, the exhaust opening in the valve-face; *J J*, the valve-stems passing through guides on the back end of the stuffing-box; *K K* shows the bonnet of the steam-chest; and *M*, the spindle which carries the right and left screws. *I* is the main valve-stem; *N*, a bracket for the purpose of holding the quadrant, *O*, in position, and preventing the cut-off from varying when it is once set. This description of valve is used on nearly all large ocean steamers.

The Wheelock Automatic Cut-Off Engine.

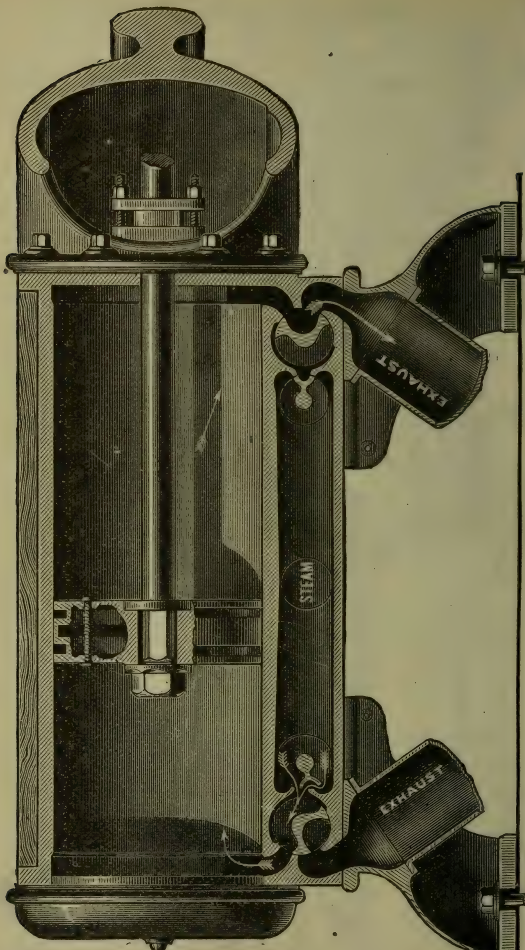
The cut on page 215 shows the cylinder, valve-gear, governor, and part of the housing of the "Wheelock Automatic Cut-Off Engine," and that on page 216 a section of the same. In general appearance, the Wheelock engine bears a close resemblance to the Corliss type, except that the absence of the cut-off valves at the top of the cylinder removes the necessity for the square corners, and that the guides, though like those of the Corliss, are parallel with the plane of vibration of the connecting-rod, and, in place of being V-shaped, are curves bored out on a line with the axis of the cylinder. This insures perfect accuracy, and prevents the possibility of the piston and cross-head getting out of line.

The valves receive their motion from an ordinary eccentric, and perform the double function of admitting and cutting off steam. Their seats are as close to the bore of the cylinder as is consistent with a proper allowance of material, thus reducing the clearance to a minimum. The valve-motion is very ingenious, effective, and simple. The cut-off is effected by tripping the valves with an arrangement which dispenses with the necessity of dash-pots, weights, or levers, as by means of lugs on the lifters coming in contact with the spring catches, which engage rock-arms on the valves, the same effect is produced. The governor is of a design peculiarly adapted to these engines, and, in consequence of its sensitiveness, holds the valve-gear under complete control, and insures a steady motion of the engine under the most varying circumstances of load and pressure.

The Wheelock engines are in very general use in the Eastern States, and seem to give satisfaction. One of them which was running in the Agricultural Department of the Centennial Exhibition, held at Philadelphia in 1876, attracted a good deal of attention, on account of its smooth and noiseless working. The most objectionable feature of these engines is the liability of the valves to become leaky.



Back View of Wheelock's Automatic Cut-Off Engine, with the Valve-Gear Unhooked.



Section of the Cylinder, Piston, Steam- and Exhaust-Valves of Wheelock's Automatic Cut-Off Engine.

Lap on the valve.—The term lap on the valve denotes the amount the edges of the valve extend over the ports when the valve is in the centre of its travel. If a valve has $\frac{7}{8}$ lap, it is understood to extend $\frac{7}{8}$ beyond the ports when placed centrally over them. The object of lap is to secure the benefit to be derived from working steam expansively. Lap on the steam side is termed outside lap, while lap on the exhaust side is termed inside lap.

Poppet- or conical-valves cannot have any lap; but the same effect is produced, as in the case of the slide-valve, by arranging the cams and lifting-toes so that the valve may close at the proper time to give the necessary degree of expansion. The lift of poppet-valves, to give an opening equal to the area of the port, is $\frac{1}{2}$ the radius or $\frac{1}{4}$ the diameter.

Lead on the valve.—The object of lead is to enable the steam to act as a cushion against the piston before it arrives at the end of the stroke, to cause it to reverse its motion easily, and also to supply steam of full pressure to the piston the instant it has passed dead-centre. It varies in different engines from $\frac{1}{32}$ to $\frac{3}{16}$, without regard to size or kind. It often, however, exceeds $\frac{3}{16}$, but perhaps very seldom; while some valves have no lead at all, others less than none, or what is termed "negative lead." The higher the speed and the more irregular the work the more lead will be required for any engine.

Loss of lead is a term employed to express the inequalities of the lead at one end of the cylinder induced by the expansion of valve-rod. It may occur, however, at both ends, through lost motion in the joints or displacement of the eccentric.

Lead on the steam end is a term applied to the amount of opening the valve has at the end of the cylinder into which the steam is entering. Lead on the exhaust end means the amount of opening the valve has on the end from which steam is escaping. The name applies alternately to each end of the cylinder.

Line and Line.—A term applied to slide-valves when they have no exhaust lead, as shown in Fig. 3, on page 204.

Valve-seat.—The flat surface which contains the ports, and on which the valve moves.

The valve-face is the working surface of the valve which moves on the valve-seat.

Valve-circle.—The term valve-circle, though sometimes used, is inappropriate, as a valve does not describe a circle. It means a circle which would have a circumference equal to the distance travelled by the valve in two strokes or one revolution. Such a circle would be smaller than that described by the centre of the eccentric, unless, as is sometimes the case, the rocker-arms were so arranged as to give a greater travel to the valve than to the eccentric.

Valve-stroke.—The travel or stroke of a slide-valve is the distance it moves on its face to give the proper opening of the port.

How to Determine the Amount of Lap and Lead on a Valve without Opening the Steam-Chest, and whether it is Equal at both Ends or not.

Open the cylinder drain-cocks and disconnect them from the drip-pipes, so that the steam may be seen and heard to issue from them. A better plan is, to open the holes made for the indicator, if there are any; at all events, open as large holes as possible; then let in a very little steam, turn the engine around by hand, and note, by the commencement and cessation of the flow of steam, just where the steam is admitted and cut-off. The point of cut-off can be most accurately ascertained by turning the engine backwards; the steam will in this case commence blowing at the same point in the stroke at which it would cease blowing when turning it forward; and, owing to the elasticity of steam, the commencement of the issue is always more clearly defined than the cessation, particularly when the issuing orifice is small. For the same reason, the point of admission can be most accurately located by turning the engine forward.

To determine the lead, having found the point of admission, make a mark on the valve-stem at a known distance from some fixed point, and another after the pin has reached the centre; this will give the lead. If the admission forward takes place when the crank-pin is exactly on the dead-centre, there is no lead. Having obtained the lead and cut-off for both ends, the travel and length of the connection being known, a diagram may be constructed similar to Figs. 1, 2, and 3, which will give the lap and port-opening.

The point of exhaust and compression cannot be determined so readily. With a small engine, in which the piston and valve are steam-tight, the points may be ascertained by blowing into the cylinder through pipes attached to the cylinder-cocks or the holes for indicator, if any. The exhaust would be indicated by the point where the air would begin to pass through into the exhaust, and the closure, by noting the point where it ceased to pass through.

But in engines of any size, especially leaky ones, the plan of blowing in with the mouth would be inapplicable. With non-condensing engines, however, much may be learned by listening to the exhaust; if the puffs occur at equal intervals, and are of equal force, good equalization may be inferred; and, if they are short, quick, and free, and are followed by a free and nearly noiseless escape of the residuary steam, the exhaust is early and ample enough. On the other hand, too late an exhaust will produce more prolonged and labored puffs. It is needless, however, to remind the reader that nothing can take the place of the indicator for determining all the conditions and adjustments of the valve, particularly its exhaust and compression, as, even when the nicest measurements and calculations are resorted to, doubts may still exist as to the truthful movements of the valve, which nothing but an application of the indicator can satisfactorily remove.

TABLE

SHOWING THE AMOUNT OF "LAP" REQUIRED FOR SLIDE-VALVES WHEN THE STEAM IS TO BE WORKED EXPANSIVELY.

When the travel of the valve is known, and the point of cut-off decided, the following table will show the amount of lap required.*

Travel of the Valve in Inches.	The Travel of the Piston when the Steam is cut off.							
	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{5}{12}$	$\frac{1}{2}$	$\frac{7}{12}$	$\frac{2}{3}$	$\frac{3}{4}$	$\frac{10}{12}$
	The required "Lap."							
2	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{8}$
$2\frac{1}{2}$	$1\frac{1}{16}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{7}{16}$
3	$1\frac{1}{4}$	$1\frac{3}{16}$	$1\frac{1}{8}$	1	$\frac{15}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{9}{16}$
$3\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{5}{16}$	$1\frac{3}{16}$	$1\frac{1}{8}$	$1\frac{1}{16}$	1	$\frac{7}{8}$	$\frac{3}{4}$
4	$1\frac{3}{4}$	$1\frac{9}{16}$	$1\frac{7}{16}$	$1\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	1	$\frac{13}{16}$
$4\frac{1}{2}$	2	$1\frac{13}{16}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{7}{8}$
5	$2\frac{1}{8}$	2	$1\frac{13}{16}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	1
$5\frac{1}{2}$	$2\frac{5}{16}$	$2\frac{3}{16}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$
6	$2\frac{1}{2}$	$2\frac{7}{16}$	$2\frac{3}{16}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{16}$
$6\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{9}{16}$	$2\frac{7}{16}$	$2\frac{3}{32}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{4}$
7	3	$2\frac{11}{16}$	$2\frac{9}{16}$	$2\frac{3}{8}$	$2\frac{3}{32}$	2	$1\frac{3}{4}$	$1\frac{7}{8}$
$7\frac{1}{2}$	$3\frac{3}{16}$	3	$2\frac{11}{16}$	$2\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{3}{32}$	$1\frac{7}{8}$	$1\frac{1}{2}$
8	$3\frac{5}{16}$	$3\frac{3}{16}$	3	$2\frac{5}{8}$	$2\frac{1}{2}$	$2\frac{3}{8}$	2	$1\frac{5}{8}$
$8\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{5}{16}$	$3\frac{3}{16}$	$2\frac{13}{16}$	$2\frac{11}{16}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{3}{4}$
9	$3\frac{13}{16}$	$3\frac{5}{8}$	$3\frac{5}{16}$	3	$2\frac{13}{16}$	$2\frac{11}{16}$	$2\frac{1}{4}$	$1\frac{7}{8}$
$9\frac{1}{2}$	4	$3\frac{13}{16}$	$3\frac{5}{8}$	$3\frac{3}{16}$	3	$2\frac{13}{16}$	$2\frac{3}{8}$	2
10	$4\frac{1}{4}$	4	$3\frac{13}{16}$	$3\frac{5}{16}$	$3\frac{3}{16}$	3	$2\frac{1}{2}$	$2\frac{1}{16}$
$10\frac{1}{2}$	$4\frac{7}{16}$	$4\frac{1}{4}$	4	$3\frac{1}{2}$	$3\frac{5}{16}$	$3\frac{1}{8}$	$2\frac{5}{8}$	$2\frac{3}{16}$
11	$4\frac{9}{16}$	$4\frac{7}{16}$	$4\frac{1}{4}$	$3\frac{5}{8}$	$3\frac{1}{2}$	$3\frac{3}{16}$	$2\frac{3}{4}$	$2\frac{1}{4}$
$11\frac{1}{2}$	$4\frac{13}{16}$	$4\frac{9}{16}$	$4\frac{7}{16}$	$3\frac{7}{8}$	$3\frac{5}{8}$	$3\frac{3}{8}$	$2\frac{7}{8}$	$2\frac{3}{8}$
12	5	$4\frac{13}{16}$	$4\frac{9}{16}$	$4\frac{1}{8}$	4	$3\frac{5}{8}$	3	$2\frac{1}{2}$

It is not advisable to cut off earlier with a single slide-valve

* If a valve has $\frac{3}{4}$ lap, it will overlap each steam-port $\frac{3}{4}$ of an inch when placed centrally over them.

than at $\frac{1}{2}$ or $\frac{5}{8}$ stroke, as otherwise the lap would be excessive and the freedom of the exhaust impaired. In locomotives and marine engines the case is different, as the cut-off may be effected at almost any point through the agency of the link.

Rule for finding the point of cut-off required to produce a given terminal from a given initial pressure.

Divide the total terminal by the total initial pressure. The quotient will be the point of cut-off in decimal parts of the stroke.

Example.—Initial pressure, 20 lbs. per sq. in. Terminal, 13 lbs., measured from a vacuum. Then 13 lbs. $\div$ 20 = ·65 of the stroke; or divide the volume of the initial by that of the terminal, the quotient will be the point of cut-off in decimal parts of the stroke.

Example.— Vol.* of 20 = 1229. Vol. of 13 = 1842. Then 1229 $\div$ 1842 = ·667 of the stroke.

Rule for finding the point of cut-off when the initial and mean pressure are known.

Add the pressure of the atmosphere to the initial and mean pressures, and divide the mean pressure by the initial. Then find in the table of multipliers, page 69, the number nearest the quotient. Find the number opposite to it in the expansion column, and divide 100 by it; the quotient will be the point of cut-off in decimal parts of the stroke.

Example.—Stroke of piston, 10 ft. Initial pressure, 10 lbs. per sq. in. Mean pressure, 8 lbs. Mean effective pressure, atmosphere added, 22·50. Initial pressure, atmosphere added, 24·5. Quotient of first divided by last, ·918. Expansion number in table opposite, ·919, which is nearest number to above quotient, 1·6; 100 $\div$ 1·6 = ·625 or $\frac{5}{8}$ of the stroke. Either of the foregoing rules will make the cut-off take place a trifle earlier than it would in practice.

Friction of Slide-Valves.

Many estimates have been made concerning the power absorbed in overcoming the friction of slide-valves, and probably on no subject has there been a greater diversity of opinion. It has been

* See Table of Volumes, pages 76 to 87.

assumed, on the one hand, that as much as one-fourth of the power of an engine is wasted, while others claim that the loss of power is merely nominal. An idea has been very generally entertained by engineers that the number of square inches in a slide-valve, and the pressure of steam in pounds per square inch, represented the total pressure on its back; or, in other words, that the pressure was equal to the pressure of steam per square inch on the back of a valve, minus the area of the steam-ports.

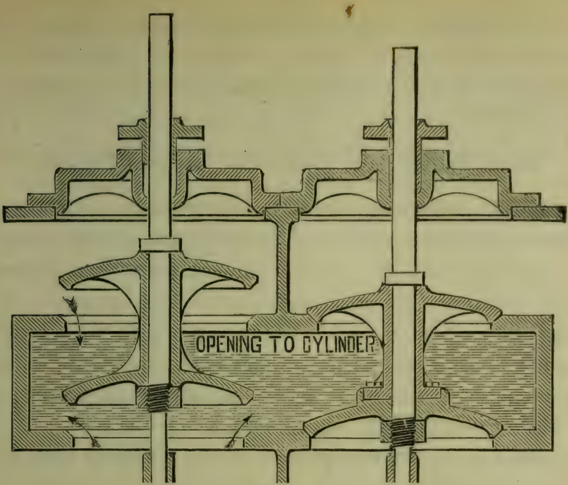
Such conclusions are erroneous, however, as the number of square inches in a slide-valve, and the pounds pressure per square inch, represent only the weight on its back, if we consider the valve as a solid block of iron, with a smooth surface resting on a smooth, solid bearing, perfectly steam-tight, in which case the steam would press on every square inch of surface with the same force as a dead weight. There is good reason to believe that such conditions are never found in a slide-valve, except in one position, viz., when the valve overlaps both ports and the engine is at rest. As soon, however, as the valve moves, the steam enters the open port, and the pressure is partially taken off that end of it.

Rule for finding the pressure on slide-valves.

Multiply the unbalanced area of the valve in inches by the pressure of steam in pounds per square inch; *add* the weight of the valve in pounds, and *multiply* the sum by 0·15.

Another rule.—*Multiply* the combined area of the bearing surface and ports in inches by the steam pressure in pounds per square inch on the back of the valve; *multiply* this product by the coefficient of friction between the two surfaces. The product will be the force required to move the valve when unbalanced.

The better the slide-valve is fitted, the more power it takes to work it; and a valve that is perfectly steam-tight on its seat, takes immensely more power to move it than if poorly fitted; because, if a valve is leaky, there is always a film of steam between the valve-face and the seat; but, when the valve is perfectly steam-tight, there is nothing to lessen the friction except the lubrication.



The above cut represents poppet,* or double-beat, valves, such as are used in connection with the Stevens' Cut-Off, or what is termed the Stevens' Front. It will be observed that the valves on the left side are open for the admission of steam, while those on the right are closed. The lift of such valves, if single, would be about $\frac{1}{4}$ of their diameter; but when they are double, as in the present case, $\frac{1}{8}$ lift would give an area equal to the opening of the steam-port. One of the greatest difficulties experienced in the working of such valves is, that, however carefully they may be fitted, their stems will expand and induce leakage in the valves when exposed to a high temperature. For this latter difficulty there appears to be no remedy.

Nevertheless such valves have their advantages, among which are, that they can be turned up, or ground on to their seats at a moderate cost, since the process of their manufacture is all lathe work; that in their working, there is no power absorbed by friction, as in the case of the slide-valve, and that they can be placed

* Puppet is the correct word, though poppet is most generally adopted by engineers.

so near the cylinder as to reduce the clearance to a minimum. Such valves, however, would not answer for high-speed engines, as at high-piston velocity, and considerable back pressure, they would not seat.

How to Set the Valves of Steam-Engines.

No definite instructions that would apply to all cases can be given for setting the valves of steam-engines. As the circumstances under which the engines and valves are employed must, to a certain extent, influence and control this operation, fast-running engines require more lead than those that run slowly. Engines doing heavy and irregular work also require more lead than those working with a uniform load. Some engines require no lead at all, while others require a great deal.

The valves of a steam-engine may be adjusted with great accuracy by an intelligent and practical engineer, providing that all the valve-gear is of correct proportions; but there are difficulties to be contended with which frustrate the efforts of the most practical mechanics, and must ever do so, unless we discover a new material for valves and valve-gear. Valves may be set with the nicest mechanical accuracy, opening and closing the ports with precision when the valves and valve-gear are cold; but when exposed to high temperatures they may be far from accurate in their travel. All metals expand with heat and contract with cold, and a valve that will give uniform lead at each end of the stroke when cold, will not, in all probability, do so when exposed to the action of the steam, as the valve and valve-rod will expand, produce a loss of lead, increase the amount of lap, and alter the conditions under which the engine was intended to work.

This change is not confined to slide-valve engines, as the stems of poppet-valves are lengthened by expansion, decreasing the lift and also the lead, and inducing a very different condition of things from what would exist if the valves could be used at the temperature at which they were adjusted. Thousands of indicator diagrams show conclusively that the behavior of valves, when exposed

to high temperature, is very different from what they are when cold. One of the best aids to correct valve-setting is a good indicator, as nothing shows the action of the steam in the cylinder so correctly as this instrument. It tells exactly when the steam goes in and out of a cylinder, because it maps down the motions of the steam as determined by the motions of the valve and piston, recording faithfully the times and pressures as they actually are.

To set a slide-valve, place the *crank* on the *dead-centre* and the valve centrally on its seat over the ports; then adjust the valve-gear to the right length, and move the *eccentric* round in the direction in which the engine is intended to run, until the proper lead is attained, as shown in Fig. 1, page 204; then turn the engine on the opposite centre, and, if the lead is exactly the same, the valve ought to travel equally on its seat, and the exhaust appear, as in Fig. 2, page 204. Any difference in the lead at either end must be equalized by lengthening or shortening the valve-gear, as the case may be.

An intelligent engineer can generally tell by observation whether engines exhaust regularly or not; as, if the steam is discharged with long or short puffs, alternately, or shows what is technically termed a long and short leg, it is evident that the valve has an earlier and a freer exhaust at one end than at the other; nevertheless, one exhaust may be heavier than the other, and yet the intervals between them may be equal. In such cases the exhaust is equal as to time, but not as to amount. The difference in amount may be caused by unequal degrees of expansion, and this in turn may be caused by unequal cut-off, or unequal clearance, or both. Such inequality cannot be cured by mere adjustment, since the lap requires to be changed; but in most cases an improvement may be effected by a compromise between equalized cut-off and exhaust, so that the effects of the inequality of both would not be noticeable.

In the case of fast-running engines, or where the exhaust has to pass through long pipes, this inequality is not easily determined from the appearance of the exhaust; but it may be done more

accurately by holding the ear close to the exhaust-pipe. This latter method may also be resorted to in the case of low-pressure engines exhausting into a condenser.

Valves and Valve-Gear.

The term valve-gear embraces all intermediate connections between the eccentric on the driving-shaft and the valves, and is applicable to all mechanical arrangements employed for working the valves of steam-engines.

The valves most generally employed for the admission of steam to the cylinders of steam-engines, are the slide, poppet, Corliss or semirotary, and rotary; plug- or piston-valves are also used, but most generally for steam-pumps. All valves, whether used for the admission or escape of steam to or from the cylinders of steam-engines, receive their motion from cams, eccentrics, or cranks; the movements of the former being indefinite as to character, and of the two latter, definite. Whatever the device employed to give motion to the valves may be termed, whether cams, eccentrics, cranks, gearing, rockers, wrist-plates, toes, lifters, trips, links, rods, levers, etc., they may be placed under the head of valve-gear.

There are engines without valves, such as the Wardwell, which was on exhibition at the Centennial Exposition at Philadelphia, and some kinds of oscillating engines, in which faces on the cylinder fit against faces on stationary steam-chests, through which the steam enters and escapes from the cylinder. Such arrangements may be called stationary valves, but they possess inherent defects, which render them useless for the most important purposes for which the steam-engine is employed.

A "releasing" valve-gear is an arrangement in which the valve is liberated from the control of its moving agent, and allowed to close in obedience to the action of a spring, weight, or other force independent of that which opened it. The agent which determines the time of release may be the governor, or it may be, and often is, some device adjustable by hand.

An automatic cut-off valve-gear is one in which the movement of the cut-off valve is so controlled by the governor, as to cut off the steam as early or as late in the stroke as may be required, to maintain the desired uniformity of speed, under variations of load and pressure.

A positive cut-off is an arrangement of valve-gear by which the expansion of the steam is effected by what is known as *lap* on the valve, the steam being cut off at the same point in each stroke, independent of load or pressure.

An "adjustable" cut-off is an arrangement of valve-gear, in which the point of cut-off can be adjusted by the hand of the engineer, outside of the steam-chest, by means of a screw, hand-wheel, or other mechanical arrangement, to meet the requirements of work and pressure. The link, in its application to the steam-engine, belongs to this class of cut-offs, as it effects the adjustment of the cut-off by means of coincident variations in the travel and angular advance using a single valve.

Riding cut-off.— A term applied to cut-off valves which ride on the back of the main steam-valve.

An independent cut-off is one in which the expansion is effected by an independent or auxiliary valve riding on the back of the main valve, and receiving its motion from an independent eccentric.

An "expansion" valve-gear is one that cuts off the supply of steam at any required point of the stroke. It embraces all the foregoing arrangements.

A "whole" stroke valve-gear is one that admits steam through the whole length of the stroke.

A "reversing" valve-gear is an arrangement employed for reversing the motion of engines. It is effected in different ways: in some cases with a single eccentric, while in others with two eccentrics, as in the case of the link; and in others, still, by means of a loose eccentric which revolves on the shaft, but is prevented from making a complete revolution by two stops so placed that one arrests it in the proper position for the forward, and the other

for the backward motion. This arrangement is peculiarly adapted to tug-boats and ferries, owing to the ease and quickness with which the engine can be reversed.

Double-beat valves are poppet-valves so arranged, that the pressure of steam is nearly equal on both sides, thus rendering the motion of the valve much easier than in the case of an ordinary single-beat valve. (See cut, page 223.)

Throttle-valves are valves located in the steam-pipe, through which steam is admitted to the steam-chest. At present their use is confined to locomotives and old-fashioned stationary engines.

Relief-valves are used on the cylinders of large engines, particularly marine, to prevent fracture of the cylinder-head and cylinder, in consequence of an accumulation of water in the latter. When a greater pressure is exerted in the cylinder than would result from the ordinary pressure of the steam, the relief-valve will open and admit of the discharge of the water, thus averting an accident. They are used on fire-engines for the purpose of preventing the hose from bursting when the escape of the water is obstructed.

Balance-valves.—Arrangements by which the weight on the back of slide-valves, induced by the pressure of the steam, is relieved by the action of the steam in the steam-chest.

Rotary-valves.—A term applied to any valve that describes a revolution in working.

Semirotary-valves.—A term applied to all valves similar to the Corliss that have a vibratory or rocking motion.

Starting-valve gear.—A mechanical arrangement employed in connection with a small engine, called the starting-engine, for moving the valves of large engines when stopping or starting.

Gridiron-valves.—A modification of the slide-valve, containing a number of openings for the steam, by which means its travel and friction are materially diminished.

Dash-pot.—An arrangement employed for closing the valves of engines of the Corliss type, and in many instances for arresting the closing movement when it is sudden and violent. The dash-

pots contain usually either water or oil, though in many instances they are cushioned with air.

Spring-levers.—Arrangements employed for closing semirotary- and poppet-valves. They are a substitute for the dash-pot, which has many advantages over them, on account of the disagreeable noise induced by their workings.

Lifters.—A term applied to the toes on the lifting-rods, which open and close the valves of steam-engines, particularly those constructed with what is called a Stevens' front.

Wrist-plate.—An arrangement employed in engines of the Corliss type for transmitting the motion of the eccentric to the valves, and in many instances for modifying their throw or movement.

Trips.—A term applied to the pawls which liberate the valves of engines having what is termed a releasing-valve gear.

Crab-claw.—A term applied to the pawls, which liberate the valve-gear of engines of the Corliss type from the influence of the eccentric, when the point of cut-off is reached.

Valves and Cocks Connected with Engines and Boilers.

The valves and cocks on a *ship's side*, in the engine, boiler-room, and hold of a *steamship*, are the injection-, main-, bilge-, discharge-, and water-service valves, and the blow-off-, scum-, and ash-cocks.

The valves on a marine engine that can be worked by hand are the stop-, safety-, slide-, throttle-, starting-, feed-, and suction-valves.

The valves that are worked by the motion of the engine are the slide, cut-off, or expansion, feed, and bilge-pump, check, and discharge valves.

The valves and pipes, through which the steam passes from the boiler to the condenser, are the steam stop-valve on the boiler-dome, the steam-pipe, the throttle-valve, the slide- or poppet-valve in the steam-chest, and the eduction-pipe between the cylinder and the condenser.

The cocks and valves through which the injection and boiler

feed-water passes in jet-condensing engines are the sea-injection cock, passing through the ship's side to the rose-plate in the condenser, from which it is drawn off by the air-pump, through the foot-valve, and delivered to the hot well, from whence the quantity necessary is drawn by the feed-pump, and forced through the check-valves into the boiler.

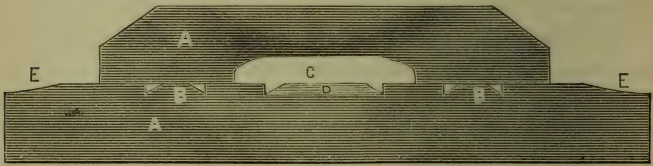


Fig. 1.

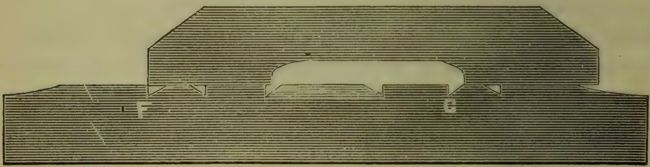


Fig. 2.

The above cuts represent a simple method of ascertaining whether a slide-valve is well proportioned or not, and whether the exhaust opens at the right time, too soon, **or too late. Disconnect** the valve from the rod or yoke, and take two parallel pieces, *A A*, about one-half inch thick and one inch wide; though the exact width or thickness is immaterial. Let one be the exact length of the valve in the direction of its travel, on which the width of the exhaust opening, *C*, in the valve-face may be marked by cutting notches with a penknife; then place the other parallel piece on the valve-seat, and mark the width of the steam-ports, *B*, and the exhaust, *D*. Then place the one representing the valve, *A*, in the centre of its travel, as shown in Fig. 1, and observe the inside and outside lap; next place it at the commencement of its stroke, as shown at *F*, Fig. 2, and observe the amount of exhaust opening.

If it should appear that the valve is well proportioned for the admission of the steam, and that the exhaust opens too late, the difficulty may be remedied by chipping out the exhaust-opening in the valve-face; or, should it be found that the exhaust opens too early, it may be obviated by inserting some pieces of brass or copper, and securing them to the valve with some small-tap-bolts, the heads of which may be riveted down; after which the pieces may be filed and scraped down to correspond to the face of the valve.

Pipes.

The principal pipes connected with marine engines and boilers are the main steam-pipe, donkey-pipe, cylinder jacket-pipe, whistle-pipe, the steam winch-pipe, ballast engine-pipe, feed-pipes, donkey feed-pipes, donkey suction-pipes, and a hot-well connection-pipe, circulating water-pipes, feed suction-pipes, air-pump discharge, bilge-discharge, bilge-suction, bilge-injection, cylinder drain-pipes, slide-jacket drain-pipes, and steam-jacket drain-pipes, blow-off- and scum-pipes, waste-steam pipe, cooling-pipe, water-service pipes.

The pipes, cocks, and valves used in connection with the locomotive are the arch-pipes, blast-pipes, connecting-pipes, oil-pipes, steam-gauge pipe, blower-pipe, feed-pipes, heater-pipes, lifting-pipe, sand-pipes, steam-pipe, throttle-pipe, blow-off cocks, check-valve, cylinder-cocks, feed-water cocks, frost-cocks, gauge-cocks, heater-cocks, mud-cock, pet-cock, safety-valve, slide-valve, stop-cock, stop-valves, and throttle-valve.

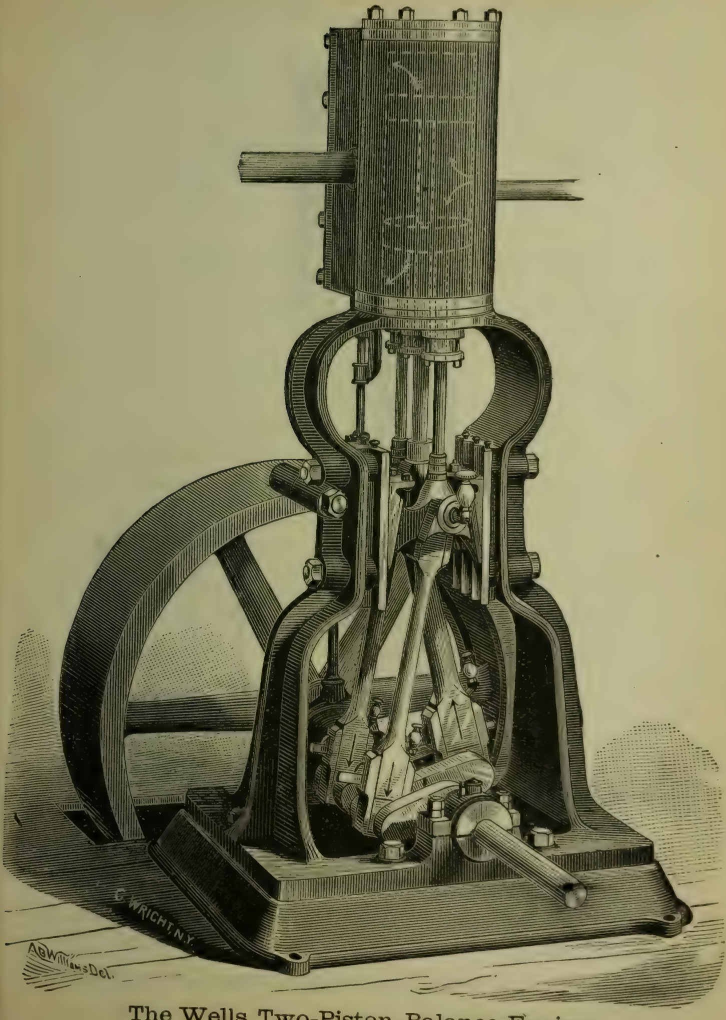
The pipes, cocks, and valves used in connection with stationary engines are the steam-pipe, exhaust-pipe, feed-water pipe, blow-off pipe, drip-pipes from cylinder, drip-pipe from heater, steam-gauge pipe, slide, poppet, or rotary steam-valves, globe-valves on steam- and water-pipes, check-valves, stop-cocks on blow-off pipe, bib-cocks, drips, etc.

Check-valves are placed on the connections between steam-boilers and the pump or injector, by which they are fed to resist the pressure from the boiler.

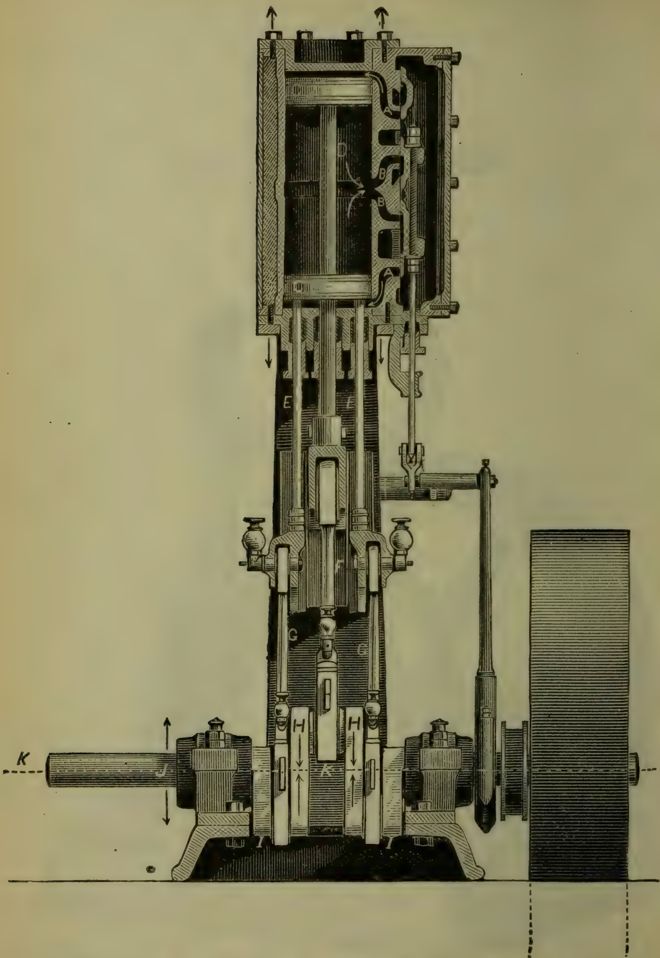
The Wells Two-Piston Balance-Engine.

The cut on opposite page represents the Wells Two-Piston Balance-Engine, which, the inventor claims, possesses features in point of efficiency and economy which place it on a par with some of the most improved engines in the country, as it may be run at a much higher velocity, and, in consequence of its greater capacity, is more efficient than any single-piston engine in use at the present day. The weight and momentum of the reciprocating parts being equal in opposite directions, the action is perfect without *lead*, which results in a great saving of steam; and as the force is applied on opposite sides of the shaft, and both cranks travel in the same direction, the thrust due to a single crank, is avoided. Moreover, because all the force of the steam on the cranks is exerted in torsion, there is no strain on the housing or foundation; hence it requires only a slight foundation. They have been frequently run at a piston speed of 1000 revolutions per minute, without any perceptible jar to the engine or vibration in the building.

It is further claimed that the advantages of high piston speed, and the benefits to be derived from expansion, are more fully realized in this engine; and that the condensation is less than it possibly can be in any single-piston engine. Besides, the weight of these engines is only about one-fourth that of ordinary engines of the same power; and, in consequence of the absence of all vibration while they are working, they can be placed in any room in a building without inconvenience or annoyance, and are peculiarly adapted to yachts and other pleasure boats. The cut on page 234 shows a section of the same engine: *A A* and *B B* designate the steam- and exhaust-ports; *C C*, the piston-heads; *D*, the middle piston-rod, which works through the middle piston-head; *E E*, the outside piston-rods; *F*, the middle connecting-rod; *G G*, the outside connecting-rods; *H H*, the middle crank-arms; *I I*, the outside crank-arms; *J*, the shaft; *K* shows the line on which the opposing strains are exerted.



The Wells Two-Piston Balance-Engine.



Section of the Wells Two-Piston Balance-Engine.

Steam is admitted simultaneously to both ends of the cylinder, and exhausted in the ordinary way by means of a slide-valve.

But as the piston is one of the most important parts of a steam-engine, and is oftener a source of annoyance and waste than any other adjunct of the machine, it is extremely doubtful if the economy of any engine can be increased by the use of two pistons. Such engines, instead of being economical, are more frequently a source of expense and annoyance.

Instructions for the Care of Steam-Engines.

Never allow an engine to become dirty, as thorough cleaning requires no great amount of labor. An engine which has always been kept clean, protected from rust and not abused in any way, is worth, when second-hand, very much more than another which has had little attention, been allowed to rust, and to take care of itself generally.

A handsomely kept engine, with all its parts clean and in good order, furnishes stronger evidence of an engineer's capabilities than a volume of written recommendations.

Never depend entirely on patent oil-cups, as they either feed too fast or not at all. There is generally too much oil wasted on engines. What is needed is a small quantity at the right time, and in the right place, and all that is not essential is wasted.

Do not allow the packing to become hard and dry in the stuffing-boxes, as under such circumstances it has a tendency to cut and flute the rods.

Never strike any part of an engine with the face of the hammer or head of a monkey-wrench, as, in consequence of their being headed with steel, they have a tendency to bruise the parts and disfigure the engine.

Never set steam-packing, cotton-waste, tops of oil-cups, or anything that is to be used round the cylinder, valves, piston-rod, or bearings of steam-engines, on the floor, as they will invariably pick up sand or grit, which injure the rubbing and revolving surfaces with which they come in contact.

When practicable, piston and valve-rod packing should be applied when the stuffing-boxes and rods are cold. The best packing is often destroyed through ignorance or want of skill.

Almost any packing may be improved by being soaked in beeswax, tallow, and black-lead, before being used.

Gum-joints that require to be frequently taken apart, should be coated with chalk before being placed between the flanges; this prevents the gum from adhering to the metal and being destroyed when the joint is taken apart. All gum-joints located in the water-space of steam-boilers should be coated with black-lead and tallow before being put together. This has the effect of preventing the sulphur of the gum from attacking the metal and destroying the surfaces.

When it becomes necessary to stop an engine with a heavy fire in the furnace, place a layer of fresh coal on the fire, shut the damper, and start the injector or pump, for the purpose of keeping up the circulation in the boiler.

Always see that the cylinder drain-cocks are open when the engine is standing still, and never close them till after starting.

Never admit the tallow to the cylinder until the engine is fairly under way, and the cylinder drain-cocks closed.

Always start an engine slowly, and allow it to come up to speed gradually.

Before starting an engine, always warm up the cylinder by admitting the steam to both ends; if a marine engine, see that everything is clear of the engines and propeller, and that the cocks and valves are all right.

Whenever an engine is stopped for any length of time, examine all its parts, for the purpose of seeing if they are in good order.

In cases of extreme heating, slack up on the keys and gibs, permit them to run loose for a time, and then take up the lost motion gradually.

Examine the piston-packing in the cylinder frequently, for the purpose of seeing that it is tight and in good order.

Keep the cylinder and steam-pipes well covered with some good

non-conductor, to counteract the cooling effect of the atmosphere.

Whenever a clicking noise is heard in the cylinder, open the cylinder drain-cocks, and allow the water to escape; then let them remain open until the cylinder works dry steam.

In giving instructions for the care and management of steam-engines, too much stress cannot be laid upon the injunction, "Keep your steam always at the same pressure," as, although all engines employed for manufacturing purposes have governors, they are not always reliable or capable of meeting the requirements of varying steam-pressures and varying loads; consequently, if the steam is, through neglect, permitted to rise above the working pressure, the engine will increase its speed, which will induce a loss of steam, as every revolution above the speed at which the engine was intended to run, and at which the machinery is geared for the manufacture of the different materials, is a waste; and every revolution the engine falls below the regular speed is a loss of production.

Piston-Rod and Valve-Rod Packing, and How to Use it.

Probably no part of the steam-engine is more frequently out of order than the piston-rod and valve-rod packing. This may be attributed to various causes, viz., such as the speed of the engine; whether it is in line or not; whether the piston leaks or not; the condition of the piston-rod; the pressure of the steam; the clearance in the cylinder, and the character or quality of the material of which the packing is composed, as well as the manner in which it is applied, and how it is treated afterwards.

If the engine is out of line, the piston-rod will crowd the packing to one side or the other at certain points of the stroke; if the piston- and valve-rod are badly fluted, the steam will escape through the grooves; if the piston-packing leaks in the cylinder, it will be impossible to keep the packing around the rod tight, in consequence of the cushioning induced by the leakage. If the distance between the piston and cylinder-head is not sufficient, the steam

will escape through the stuffing-box as the engine approaches the centres; if the rings of the material are cut too long, they will not, when screwed up, hug the rod, and, as a result, leakage will occur; if too short, the steam will insinuate itself between them, and cause leakage; if the material is not of the proper size to fill the cavity between the rod and the box, it will leak, however tightly it may be screwed up; if the packing is screwed up too tight at first, the heat induced by the friction will soon destroy its elasticity, and leakage will be the result; if the engine runs at a very high speed, the packing will deteriorate faster than if the speed is moderate; and if the pressure is high, the temperature due to the pressure will have a tendency to destroy the packing. Another cause, and indeed one of the main causes which induce leaking around piston- and valve-rods, is the want of depth of the stuffing-boxes of some engines, which will not receive more than two rings of packing; as a result, they are continually leaking around the rod, whereas, if the box is sufficiently deep to admit of four rings, the leakage nuisance would be obviated.

A great variety of materials is in use for packing purposes, soap-stone, paper, india-rubber, asbestos, tin-foil, webbing, wire-cloth, metallic packing, etc., each of which possesses merit peculiar to itself, but, like governors, and many other important adjuncts of the steam-engine, not one of them was ever known to answer every place, or give satisfaction under all circumstances. This arises from the fact that our investigation has not been such up to the present time, on this subject, as to enable us to decide which material will give the most satisfaction under the most varying circumstances; besides, the best material may be rendered useless in a comparatively short time through ignorance, while an inferior quality may render good service by being intelligently treated.

The following instructions may be of use to those who have not had much experience in packing piston- and valve-rods: Insert as much packing into the box as will just allow the gland to enter; then screw it up solid; after which the nuts should be slacked for the purpose of allowing the packing to swell when exposed to the

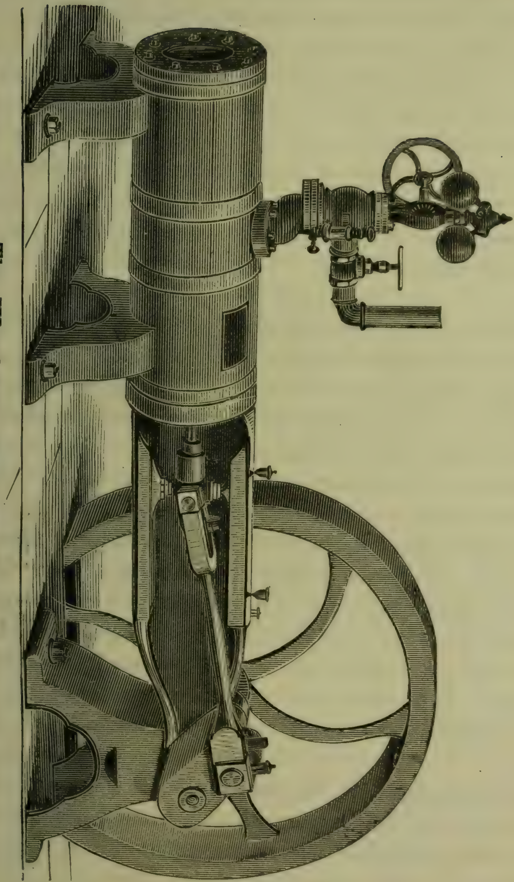
steam; if leakage occurs, screw them up gradually and evenly, until it stops. If the leakage is excessive, after a sufficient quantity is inserted in the box, do not continue to screw it up, as the heat of the rod will soon destroy the packing. It is always better to stop the engine, if practicable, remove two or three pieces, and replace them again in opposite positions, when in all probability the leakage will cease. Never use any old file or any rough instrument to remove the packing from the boxes, as they have a tendency to abrade or flute the rods, and cause leakage. Every engineer and steam-user should provide himself with suitable tools for removing the old packing from the boxes and inserting the new. To find the proper diameter of the packing for any stuffing-box: Measure the diameter of the rod, and also the gland or stem of the stuffing-box, and half the difference between the two will be the proper size of the packing.

Numerous attempts have been made at different times to substitute a metallic piston- and valve-rod packing for the various fibrous packings now in use which would be more durable, and at the same time involve no more cost; but up to the present time none of these attempts have been crowned with success. This may be attributed to various causes, such as the condition of the piston-rod, whether it is fluted or not; whether the engine is in line or not; the condition of the steam-packing in the cylinder; the depth of the stuffing-box, whether it is leaky or not; the clearance space between the piston- and cylinder-heads when the crank is at the centre; the amount of back pressure; the difficulty of manufacturing the metallic packing in sizes to meet all the vagaries of that class of steam-engine builders who pay no attention to good proportions, and who make the stuffing-boxes odd sizes; the condition and shape of a stuffing-box to which it has to be applied; the ignorance displayed in its care and management, as well as a disposition on the part of those who have it in charge to cry down every new innovation in steam engineering, and to ridicule every adjunct of the steam-engine and boiler that requires any special attention, however great a safeguard or economizer it may be.

Wardwell's High-Pressure, Valveless Engine.

The cut on page 241 represents Wardwell's Valveless Engine. As will be observed, it is a horizontal engine, with one end of a girder frame bolted to and supporting the cylinder, and the other supporting the pillow-block. The pillow-block brasses are provided with side adjustment wedges, operated from the top face of the cap by bolts and nuts. The cross-head has V-shaped bearings, top and bottom, with a wrist-pin providing journal-bearings for the fork end of the connecting-rod. The straps at these ends of the rod are provided with the ordinary gibs and keys. At the crank-pin end, however, the strap is secured to the rod by a bolt passing through the strap, the key merely serving to adjust the brasses. The piston passes a working fit through the cross-head, being secured at each end by jamb-nuts, by which arrangement any lateral play of the piston-rod in the cross-head is prevented; but at the same time the rod rotates in the latter. To the extreme end of the piston-rod, after it has passed through the cross-head, there is keyed fast a section of a bevel-wheel containing 5 teeth, which gears into another containing 4 teeth; this latter section being bolted fast to the inside of one of the fork-arms of the connecting-rod; the outside arm being selected as affording the best advantages for adjustment. When the connecting-rod is attached to the crank- and cross-head, and steam admitted to the cylinder, a semirotary movement takes place in regular order, and as the stroke proceeds, the steam passages are so arranged that steam can be admitted, cut off, and exhausted at any desired point of the stroke. It is obvious, however, that to accomplish this the piston-head in the cylinder must be extra long in proportion to the stroke.

The piston is solid, similar to a plunger, and is a neat working fit in the bore of the cylinder. The wear is provided for by the insertion at each end of the piston-head of ordinary spring packing-rings; and to take up wear and prevent leakage from one port to the other, a straight, longitudinal, spring packing-piece is placed



The Wardwell Valveless Engine.

between the steam passages in the piston-head, thus preventing the escape from one port to the other. The steam-port is in the centre of the cylinder, and on top. The steam passages in the piston-head commence near one end, and run along the circumferential surface, in a longitudinal but curved line, so that the passage will remain full open to the cylinder steam-port, notwithstanding the rotary motion of the piston. At such part of the stroke, however, at the point at which the steam is to be cut off, the steam passage turns at an angle, and runs round nearly one-half the perimeter of the piston-head, so that the rotary motion of the piston during the remainder of the stroke is insufficient to permit any communication between the cylinder-port and piston passages. So soon as the piston-head steam passage turns the angle above noted, the longitudinal movement of the piston-head past the cylinder steam-port cuts off the supply of steam, and the remainder of the piston-stroke is performed by expansion. The circumferential direction of the passage above referred to serves another purpose than acting as a cut-off, in that it enables the same passage to be used to convey the steam to the cylinder exhaust-port. After the steam passage has taken the circumferential direction referred to, it continues longitudinally to the end of the piston-head; the steam passage, while isolated from the cylinder steam-port, comes into open communication with the cylinder exhaust-port, and that stroke of the engine is completed. For the return-stroke, a similarly arranged passage is provided in the piston-head, and hence the piston requires but two passages, each of which operates alternately, as induction and eduction passages.

There were three of this description of engines on exhibition at the Centennial Exposition, which attracted considerable attention, in consequence of the arrangement for admitting and exhausting steam being entirely different from anything heretofore employed. Such engines possess no practical value, their chief interest consists in the novelty of the arrangement.

Lubricants.

To understand the quantity of oil required for steam-cylinders, slide-valves, and the reciprocating or revolving parts of steam-engines, we should first know what its objects are. The object of oil is to diminish friction, by interposing a thin film between the sliding or revolving surfaces. To insure perfect lubrication, the surfaces must be kept coated at all times, under all pressures and velocities. In steam-engines there is a sliding and rotating friction, and it is very doubtful if any one kind of oil is perfectly suited to both. Oil has no tendency to improve the character of a bearing; its functions being simply to keep them apart, prevent heat, and diminish friction.

Temperature exerts a very important influence over any lubricant. A thin oil has a tendency to run off too fast, while a thick one is not sure to flow. Tallow, and all other thick and greasy compounds, are exposed to the same objection, as the bearing generally gets hot before the lubricant begins to flow. Besides, what may be called a good lubricant, one that adheres to the rubbing surfaces under ordinary circumstances, may not be equally well adapted to all conditions, as the area of the bearing surfaces varies with the size of the journals, and the form of the boxes, which causes a difference in the velocity of rotation. From this, it follows, that a lubricant that would be retained between the frictional surfaces under a light load, would be entirely pressed out under a heavy one.

The quantity of lubrication that the cylinders and slide-valves of any engine require, depends on the condition of the engine, the amount of work it is performing, and on the pressure and temperature of the steam. If the load is light, the pressure low, and the engine in good order, very little lubrication is necessary; but if the pressure and speed are high, and the engine is working up to its full capacity, the cylinder and valves will require to be frequently lubricated. But in no case should an unnecessary quantity be used, as it is likely to produce a greater evil than the

one it was intended to remedy. A person having charge of steam machinery should understand the work each part has to perform, the speed at which it runs, and the weight it has to sustain. Crank-pins and main-bearings require to be frequently oiled; but the condition of the bearing will determine the quantity of lubrication needed. What is needed in any case is a few drops of good oil applied often. It may be safely said that five times the quantity of lubrication is used on the revolving and rubbing surfaces, and in the valves and cylinder of steam-engines, which is actually necessary.

According to the general impression, grease or animal oil is a preserver of metal; but experience has shown that it is more frequently a destroyer, especially of the cylinders, pistons, and valves of steam-engines. The reason of this is, that vegetable and animal fats and oils contain stearic, megaric, and oleic acids, which, when subjected to the heat of high pressure steam, that frees them from their base, attack the metal and destroy it. This applies as well to oils of vegetable as to oils of animal origin, as fish or sperm oil. On removing the heads of steam-cylinders and the bonnets of steam-chests, the cylinders, pistons, and steam-chests frequently show evidence of corrosion, which differs entirely from that of ordinary wear, and which persons unacquainted with the nature and effect of the oil and grease they have been using, are puzzled to account for. Oils derived from petroleum contain no oxygen, cannot form acid, and therefore do not attack metal. The proof of this may be found in the fact, that such oils are used in surgical operations, and for cuts, bruises, and abrasions, with good effect. Oils from petroleum are produced for nearly every mechanical process, as well as for the cylinders of steam-engines, for which latter purpose animal oils were considered indispensable.

At a recent meeting of the Railway Master Mechanics' Association, at St. Louis, a report was presented by the committee on lubricants, which embodied the result of a series of experiments made for the purpose of testing the lubricating qualities of different kinds of oil. In making the test, 56 drops of each variety

of oil were put into a dynamometer, which was run at 35 miles an hour, until the temperature was raised from 60° to 200° Fah. The exact number of revolutions necessary to produce this change of temperature was noted in each case, and is given in the last column of the following table.

Description of Oil.	Cost per Gallon.	Average Revolutions.
Castor Oil	$1.25	12·946
Paraffine	.28	11·685
Mecca (black)	.45	9·982
Manufactured "*A*"	.35	9·653
" " "*B*"	.90	9·394
" " "*C*"	.25	9·187
Neat's-foot	.85	8·277
W. Virginia	.26	7·915
Sperm	1.75	7·912
Tallow	.70	7·794
No. 1 Lard	.70	7·377
Manufactured "*D*"	.25	6·999
" " "*E*"	.85	6·798
" " "*F*"	.20	6·121
W. Virginia (reduced) . . .	.20	4·770
Grafton (treated)	.20	4·215

It has lately been demonstrated that natural petroleum oils, when thoroughly freed from grit, are for many purposes as good, if not better, than sperm, with the advantage of being much cheaper; but they are objectionable in consequence of their liability to stain bright work or finished machinery.

It is not by any means uncommon to see ignorant and inexperienced persons who have charge of steam-engines pouring oil on cross-head guides and piston-rods every five minutes during the day. This is immediately rubbed off by the shoes or the piston-rod packing, without rendering any service, which is a wilful waste of the necessary supplies in their charge, and has a tendency to lessen the profits of the establishment.

Questions:

THE ANSWERS TO WHICH WILL BE FOUND IN THE TEXT.

What are the objects and functions of the bed-plate as a part of the steam-engine?

Give the rule for finding the necessary strength of a bed-plate for any given speed and pressure.

State the rule for finding the proper thickness of a steam-cylinder of any diameter.

Give the rule for finding the diameter of a cylinder for any given horse-power.

Give the rule for finding the cubic contents of a steam-cylinder of any given diameter.

State the rule for finding the quantity of steam that any engine will use at each stroke of the piston.

Give the necessary strength of cylinder-head bolts.

What are the objects and functions of the pistons of steam-engines, and what qualities should they possess?

Give the proportions of piston-rods for condensing and non-condensing engines, according to the best modern practice.

Give the units of horse-power for various piston speeds.

Give the proportions of steam- and exhaust-pipes according to the best modern practice.

What proportion should the diameter of the valve-rod bear to that of the cylinder?

Give the proper length and width of the cross-head bearings.

What is the meaning of the term eccentric?

What are the functions of the crank, and what change of motion does it induce?

Explain the cause of the variation of the piston in the cylinders of steam-engines when their cranks are at half-stroke.

Give the rule for finding the position of the piston in the cylinder when the crank is at half-stroke.

Is there any loss of power incurred in the employment of a crank as a mechanical device for converting reciprocating into rotary motion?

Give the rule for finding the necessary proportions of crank-pins for any engine.

Give the proportion of the crank-shaft and main-bearings according to the best modern practice.

Give the proper proportions of gibs, keys, and straps for any engine.

Why is the strap thicker at the *slot* than at the part which encircles the box?

What are the functions of the link?

What is the meaning of the term "radius of the link"?

Describe the mechanism of the various links employed as a reversing-gear for steam-engines.

What is the object of the fly-wheel?

Give the rule for finding the proper weight of fly-wheel for any engine, speed and pressure being given.

What are the functions of the steam-engine governor?

Give the rule for finding the proper size of governor-pulleys.

Give the most approved method of counterbalancing the reciprocating parts of steam-engines.

What are the most common causes of heating in the journals of steam-engines?

What are the uses and functions of the slide-valve?

Explain the advantages and disadvantages of the slide-valve as contrasted with those of other forms.

Explain the action of poppet- or conical-valves.

What are the meanings of the terms *lap* and *lead* on the valve?

What is the meaning of the term *loss of lead?*

What is meant by the terms valve-seat and valve-face?

Explain the meaning of the terms valve-circle and valve-stroke.

How would you proceed to ascertain the amount of lap and lead on a slide-valve without opening the steam-chest?

Give the meaning of the term cut-off, and the amount of lap required to cut-off at different points of the stroke.

State the most economical point in the stroke at which to cut off the steam, and demonstrate it by an example.

What is meant by friction when applied to slide-valves?

How would you proceed to set the valves of a steam-engine?

Is the friction of a perfectly fitting slide-valve more or less than that of an imperfectly fitting one?

Give the names of the different valves and valve-gear employed for the admission and escape of the steam to and from the cylinders of steam-engines.

Give the technical terms as applied to the valve-gear of steam-engines.

Give the names of the various valves and cocks in use on different steam-engines and boilers.

Give the names of the various pipes in connection with different kinds of steam-engines.

Explain the meaning of the term valve-gear.

In what condition should an engine be kept?

What is the best evidence of an engineer's capabilities?

What dependence should be placed on patent oil-cups?

In what condition should the packing in the piston- and valve-rod boxes be kept?

What is the objection to striking any part of an engine with the face of a hammer or head of a monkey-wrench?

What is the objection to placing piston-rod packing or cotton waste on the floor?

When should piston- and valve-rod packing be applied?

In what manner may piston- and valve-rod packing be improved?

How should gum-joints be treated, which must of necessity be frequently broken?

What precaution should an engineer take, when it becomes necessary to stop an engine with a heavy fire in the furnace?

How should the cylinder drip-cocks be kept, when an engine is stopped?

When should the tallow or any other lubricant be admitted to the cylinder?

How should an engine be started?

What precaution should an engineer take before starting an engine?

What course should an engineer pursue when it becomes necessary to stop for any length of time?

What course should be adopted in case of extreme heating in any of the revolving parts of an engine?

How should the piston-packing in the cylinder be treated?

What are the best means of protecting the cylinder and steam-pipes from the effects of the atmosphere in order to diminish radiation and condensation?

What course should be adopted when a clicking sound is heard in the cylinder?

Why is it very important that the steam pressure should be kept uniform?

Give the reasons why the piston- and valve-rod packing is so frequently out of order.

Explain the best method of using piston- and valve-rod packing.

What is the best course to adopt when excessive leakage occurs?

How should piston- and valve-rod packing be kept?

Give the rule for finding the right diameter of packing for any stuffing-box.

What is the object of lubrication?

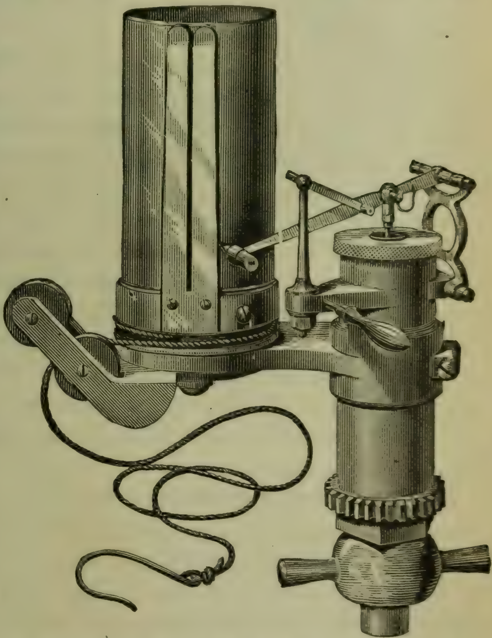
What effect has temperature on lubricants?

What conditions influence the amount of lubrication required for any engine?

PART FOURTH.

The Steam-Engine Indicator: Its Invention and Improvement.

Perhaps no device, in the entire range of mechanical inventions, has aided so much in developing and perfecting the steam-engine as the indicator. This arises from the fact that no other invention yet brought forward pertaining to the science of steam engineering can read the inner workings of a steam-engine, point them out with unerring accuracy, and discover the sources of waste in it. Consequently, its importance cannot be too highly estimated, and its use too much encouraged and extended in all classes of steam-engines.



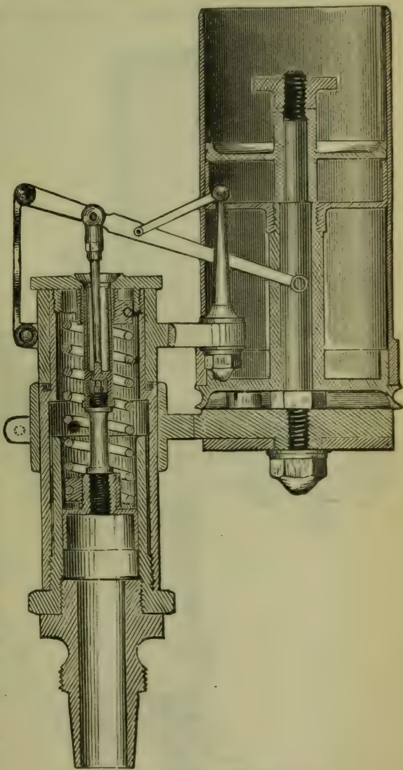
The steam-engine indicator is said to have been invented by James Watt, which is rather doubtful; and, as Watt received credit for many things he never invented, it is not to be wondered at that the invention of the indicator has been attributed to him. Be that as it may, Watt's indicator, though very im-

perfect, answered for engines travelling at a piston speed of about 150 feet per minute, and for pressures averaging 7 lbs. above atmosphere, which he thought was the fastest speed and the highest pressure that would ever be needed. But experience soon demonstrated that the highest economy was attained with high piston speeds and correspondingly high pressures, and, as a result, Watt's indicator proved to be unsuitable for these conditions. The requirements of such an instrument were more fully appreciated by McNought, of Glasgow. The world is more indebted to him for improvements in the steam-engine indicator than to any one previous to his time.

The indicator was further improved by Hopkinson, Stillman, and others; but these improvements were not in the mechanical design or arrangement of its working parts, but rather in the accuracy and refinement of the workmanship employed in its construction, as the mechanical principles embodied in the Watt indicator were continued in them all. They consisted of an upright cylinder, into which a piston was accurately fitted. To the piston-rod a spiral spring was attached, to resist the steam and the vacuum when acting against it. The pencil was also attached to the piston-rod; the result of which was that the piston, piston-rod, and spring had the same movements as the pencil. With such instruments the vibration of the piston was so great as to render them totally unreliable with fast running engines, or when steam was worked expansively.

Gooch was the first inventor that gave the pencil a greater range of movement than the piston. In his instrument the cylinder was placed horizontally, and when its piston was subjected to pressure it compressed two elliptic springs. The top of his piston-rod was connected to the short arm of a lever, to the long arm of which the pencil was attached, thus giving considerably more motion than could be obtained by any former instrument. The pencil moved in the arc of a circle instead of a straight line. The diagram was traced on a web of paper while it was unwound

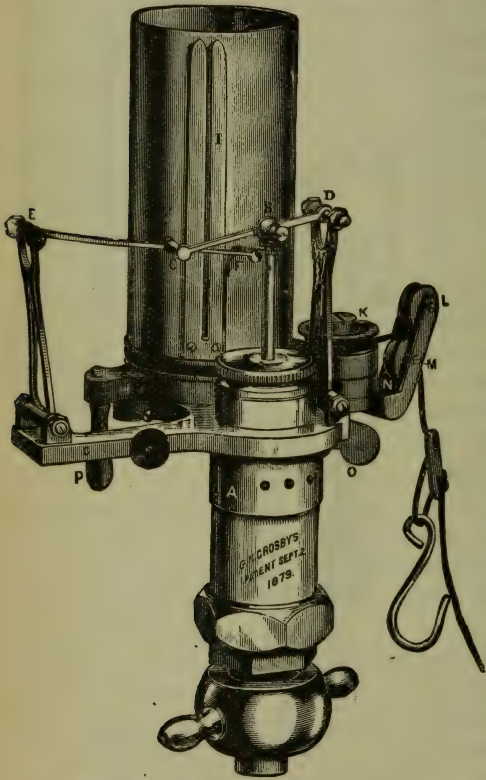
from one drum and wound upon another. This arrangement admitted of a succession of diagrams being taken without any intermediate manipulation of the instrument. The communication between the indicator and the steam-cylinder was closed by a slide-valve instead of a cock. But as the principle of working steam expansively became almost universal, an instrument more reliable than any of these previously mentioned became a necessity of the times, and such was found in the Richards' Indicator. In this instrument the following construction and proportions have been adopted, and adhered to from the first. The area of the piston is $\frac{1}{2}$ a square inch, the diameter of which is very nearly $\frac{8}{10}$ of an inch, or, more exactly, ·79 inch. The length of the long arm of the lever, to which the rod of the piston is attached, is 3 inches, and the distance from the pivot of the lever to the point of attachment of the piston is $\frac{3}{4}$ of an inch, thus giving the free end of the lever, and with it the pencil, four times the movement of the piston. The secondary lever is equal in length to the first, and the link which connects the two, and which carries the pencil at its centre, is $1\cdot\frac{7}{8}$ inches long. These propor-



Section of the Indicator.

tions give a practically straight pencil movement for a distance of $2\frac{1}{2}$ inches.

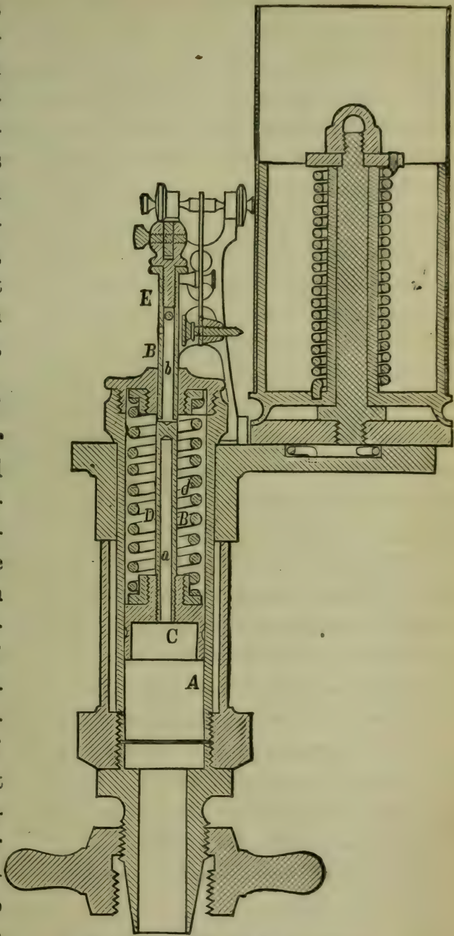
The indicator was further improved by Harris Tabor, (cuts of whose instrument may be seen on pages 260 and 261;) but more recent improvements made in the indicator have been effected by George H. Crosby, a mechanical engineer of Boston, Mass. It has apparently been the aim of Mr. Crosby to avoid unnecessary weight in the reciprocating parts, to insure correctness of action, and to so simplify the method of manipulating the instrument as to bring it within the understanding of engineers of limited education and persons of ordinary intelligence. In these objects he seems to have been partially successful,



Crosby's Improved Indicator.

as the Crosby Indicator is an improvement, in some respects, on other devices of the kind in use; as it is reliable in its recordings, whether employed for taking diagrams from automatic cut-off, throttling, simple, compound, fast- or slow-run-

ning engines; as it is free from some objectionable features in other instruments, which render the diagrams taken by them erroneous. In the conception of this instrument, the inventor seems to have predetermined many of the circumstances, emergencies, and requirements that might possibly arise in the use of the indicator, and provided for them.



Section of Crosby's Indicator.

The advantages of the Crosby Indicator are, that the parallel motion is not a geometrical approximate imitation of, but a true motion; that the motion of the pencil is a uniform multiplication of the piston of the indicator, and is solely controlled by the motion of the piston-rod; and that there are no guiding-slots, either straight or curved, to induce friction; that there is no compensating arm jointed to any fixed point, as in other indicators; that the pencil is located close to the piston-rod, instead of projecting several inches to one side,

as in other cases; that an air-chamber or jacket surrounds the steam-cylinder instead of a steam-jacket, as in other instances; that the piston-rod is hollow instead of solid, and that it is solidly united to the piston, thus requiring no joints below the cap, which obviates the possibility of corrosion by the action of the steam or moisture; that there is no link or connecting-bar between the head of the piston-rod and lever to cause friction or inaccuracy of motion; that the cylinder, piston, and piston-rod are automatically oiled by a self-lubricating device, thus removing the possibility of friction, which always induces error in the recordings of the indicator, thus rendering the diagram deceptive even to experts; that, wherever possible, every joint is made with steel pivots instead of journals, as is the case in other instruments; that the mechanism for adjusting each instrument is so arranged that it may be used either left- or right-handed, as the case may be, in order that diagrams may be taken from either end of the cylinder without the necessity of two indicators; that means are provided for adjusting the distance that the paper shall move towards the pencil, so that a hair-line can be drawn without friction; that the reduction in weight in the piston and hollow piston-rod and the refinement of workmanship in the levers and joints, render the reciprocating parts so extremely light that momentum and friction are reduced to a minimum; that it is more easily adjusted and operated than any other instrument of the kind ever heretofore invented, thus dispensing with the necessity of experts, and that diagrams may be taken from each end of a steam-cylinder without the least difficulty, even by engineers of ordinary intelligence and limited experience, from engines running at the highest practicable piston speed.

They are manufactured by the Crosby Steam Gauge and Valve Co., Boston, Mass.

The Thompson Improved Indicator.

The Thompson Indicator, see page 251, improved and patented by J. W. Thompson, of Salem, O., is the only instrument now in

use that can be used on very high-speed engines with success; and it works equally as well on slowly as quickly running engines. It will give correct results under any attainable speed of an engine or locomotive.

The adoption of high-piston speed of stationary and locomotive engines has created a demand for an indicator that will take cards at a very high speed, say three hundred revolutions per minute, or even more.

It will be observed that Mr. Thompson's improvement mainly consists in reducing the weight of the parallel motion, by lessening the number of vibrating pieces, thereby decreasing the tendency to make wavy lines in both steam and expansion. By this arrangement, the instrument is lighter and more compact,—qualities which will be fully appreciated by all intelligent engineers.

The Thompson Indicator has taken the first premium wherever exhibited in competition with other indicators.

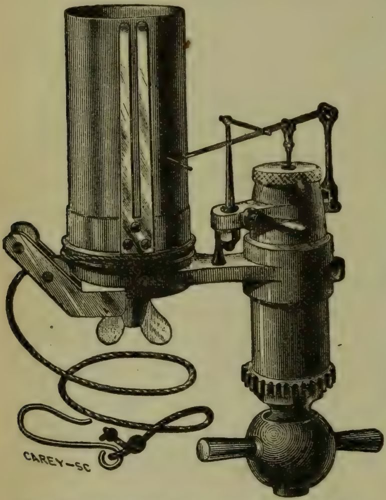
About two years since, the United States Government being desirous of ascertaining which of the various patterns of indicators in use was the most efficient, with a view to its adoption as the *Standard* for naval service, Engineer-in-Chief Wm. H. Shock, U. S. N., Chief of the Bureau of Steam Engineering of the Navy Department, issued an order directing the Commandant of the Boston Navy Yard to appoint a board for this purpose. The board consisted of three officers of the Engineer Corps, who reported unanimously in favor of the Thompson Indicator, and it was subsequently adopted by the United States Navy Department as the Standard Indicator.

The Thompson Indicator is also in use by all the principal Institutes of Technology throughout the country.

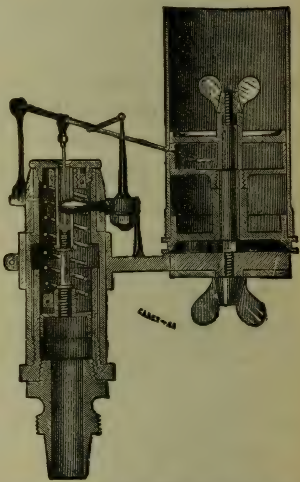
By an ingenious device, invented by Mr. Thompson, cards can be taken with this instrument at a pressure as high as five hundred pounds to the square inch.

The Thompson Indicator is manufactured by the American Steam-Gauge Company, Boston, Mass., who have been eminently successful in the manufacturing of first-class instruments for nearly

thirty years, and who, seeing the superiority of the Thompson over any other make, negotiated with Mr. Thompson for the sole manufacture and sale of same. The cut on page 251 shows the indicator as it was made when the manufacturers received it from Mr. Thompson. But knowing that the principle of the Thompson was far superior to any other indicator in the market, and being desirous of keeping far in advance of all other manufacturers of indicators, the manufacturers have made some very important improvements in the indicator, so that it is now known as the Thompson Improved Indicator, as shown by the following cuts.



The Thompson Improved Indicator.


Section of the Thompson Improved Indicator.

The important improvements consist in lightening the moving parts, substituting steel screws in place of taper-pins, using a very light steel link instead of a large brass one, reducing the weight of the pencil-lever, also weight of square in trunk of piston and lock-nut on end of spindle, and increasing the bearing on connection of parallel motion. By shortening the length and reducing

the actual weight of the paper cylinder just one-half, and by shortening the bearing on spindle so that it now carries the drum-spring nearer the base, they have reduced the momentum of the paper cylinder to a very small amount. All of these improvements have lessened the amount of friction, which was heretofore very small, but is now reduced to a minimum, and the Thompson Improved Indicator as it is now made is without a peer, and is the standard Indicator in Europe and this country, and has been adopted by the most eminent engineers in both countries.

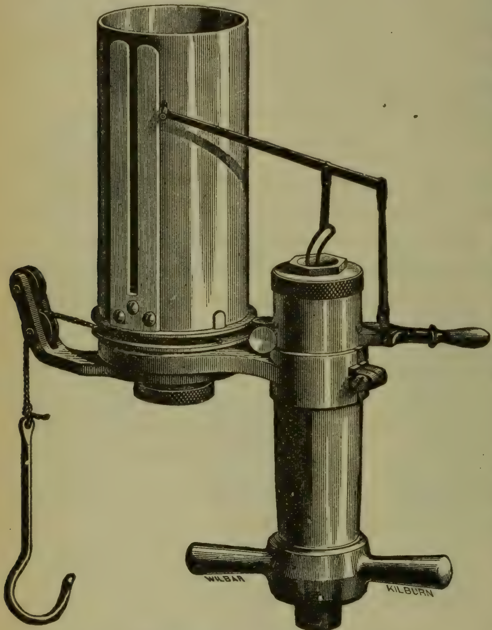
Some of the advantages of Thompson's Improved Indicator are: that it is handsome in design, and convenient and simple in arrangement; that cards can be taken at a pressure as high as 500 ℔s. to the square inch; that they are easily adjusted and operated; that all the moving parts have been lightened, which is a consideration of great importance, especially for engines travelling at a high rate of piston speed; that it is made of materials carefully selected and accurately fitted, thus insuring durability; and that they are adapted to all purposes for which such instruments are employed.

Every Indicator is tested at the works by being attached to an engine before being sent out, and tried under different pressures, which insures satisfaction in the working; so that it is sure to meet all the requirements for which it is intended.

Tabor's Indicator.

The cuts on pages 260, 261, represent Tabor's Steam-Engine Indicator. — As will be observed, its most striking features are its parallel motion and the plainness of its cylinder. The piston has a single capillary packing groove, and its whole action is remarkably nice. The springs, both as to range and general structure, are similar to those in the Richards and Thompson Indicator. It will be noticed that the piston-rod, which is jointed to the piston and the pencil-lever, is slotted; this slot is curved, and works over a guide-roller set in the cylinder-cap. The rear end of the

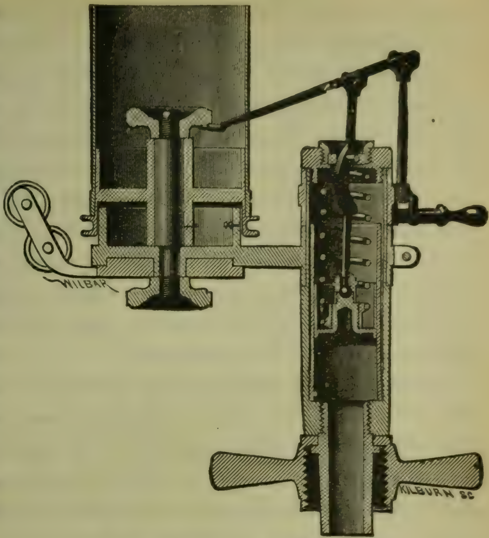
pencil-lever is pivoted to the radius link. The slot-curve is that peculiar curve which would be described by the guide-roller as a scribing point while the pencil is being moved in a true line; this, it is claimed, insures a correct parallel motion to the pencil. The guide-roller is journalled in a free collar held in the cylinder-cap, which allows all the moving parts to revolve freely, as the pencil is brought in contact with the paper.



Tabor's Indicator.

The paper drum revolves on a steel spindle, upon which the bottom nut is screwed; the nut inside the drum is simply a milled head firmly screwed on the upper end of the spindle. The recoil-spring is seated in a cup on the bracket, the outer end being fixed to it, while the inner end is hooked by the hub on the drum-base. A stop-block on the cup, engaging with a lug in the drum-base, forms the stop for the recoil motion. If the spindle be slacked somewhat. the drum-base may be revolved over the stop-block, and more or less tension given to the recoil-spring. By simply unscrewing the cylinder-cap, the whole motion work may be removed in one piece. The pencil-lever, piston-rod, and radius-link, are all of steel,

spring tempered; the small number of moving parts, and their lightness, reduce the error of momentum that exists in instruments of heavier parts, which is frequently a source of uncertainty. The whole instrument is very light, the design simple, and the workmanship neatly done. It is claimed that the diagrams produced are very good.



Section of Tabor's Indicator.

Functions of the Indicator.

The function of the indicator is to automatically trace out on paper a diagram that will graphically represent the pressure of the steam in the cylinder of the engine to which it is attached, with all its variations during both forward and return strokes of the piston. It enables those who use or have charge of steam-engines, to ascertain the condition of the parts of the engine subject to the direct action of the steam, and to what advantage the steam is applied; whether the valves are properly designed and accurately set, and if the steam-passages or ports are of the proper size to receive and discharge the steam in time to produce the best effect; what pressure of steam there is upon the piston at every position in the cylinder, as well as its average during the stroke; what is the value of the vacuum acting upon the piston of a condensing engine in all its positions in the cylinder, and what is its

average; whether the exhaust passages from the cylinder are sufficiently large to give free exit to the steam, and, if not, what percentage of power is lost in forcibly expelling it; the actual consumption of steam in giving motion to the engine, and also what additional steam is used in giving motion to the shafting and millwark, the paddle-wheel or screw-propeller; and also what power is required to move the machinery, or any part of it.

In manufacturing establishments where power is let to tenants it will show how much is consumed by each, and it will also demonstrate the degree of economy in using steam at different pressures, the benefits of expansion, and the relative efficiency of different kinds of expansion-gear.

Indicator cards are of great value, as they demonstrate the initial, mean effective, and terminal pressures, the back pressure, the cushion, whether by compression or lead; the point of cut-off, and the relative economy of different engines, aside from leakage and condensation. It may be applied not only to steam-engines, but to those driven by compressed air, or any vapor or fluid, as well as to the cylinders of air-pumps, air-compressors, blast-engines, etc. The diagram produced is the joint production of two movements, viz., a vertical movement of the marking point due to the pressure of the steam acting on the piston of the instrument, in opposition to the force of a spring of known strength, and a horizontal movement of the paper, as the drum, on which it is placed, makes partial rotations to and fro coincident with the movement of the piston. Hence, when the pencil is held in contact with the paper during one revolution of the engine, both will arrive at the point from which they started at the same moment, and a closed figure will be the result, except when a great change in the load and pressure occurs during the stroke in which the diagram was taken.

The value of indicator diagrams is that they show what proportion of the boiler pressure is contained in the cylinder; how early in the stroke the highest pressure is reached; how well it is maintained; at what point and at what pressure the steam is cut off;

whether it is cut off sharply, or in what degree it is wire-drawn; at what point, and at what pressure it is released; whether it is freely discharged, or what proportion of it (in excess of the atmosphere or the vacuum in the condenser, according as the engine is condensing or non-condensing) remains to exert a counter or back pressure; whether, before the commencement of the stroke, there is any compression of the vapor remaining in the cylinder, and if so, at what point in the stroke it commences, and to how high a pressure it rises. The foregoing particulars can only be learned by observation, though a scale, corresponding with the spring used, is needed to measure the pressures, and to locate the exact events in the stroke. The points to be observed in estimating diagrams are, the mean or average pressure; the total mean, or the mean effective pressure; the indicated horse-power, I. H. P., and the theoretical water consumption. The indicator shows the pressure at each and every point in the stroke; to represent this faithfully is its sole office. The causes which determine the form of the figure must be determined by the engineer.

Technical Terms Used in Connection with the Employment of the Indicator.

The term Adiabatic literally means no transmission. As applied to an expansion curve, it means that it correctly represents at all points the pressure due both to the volume and the temperature, just as if no transmission of heat to or from it had taken place.

Admission.—This term is applied to the induction of the steam into the cylinder when the valve opens at the commencement of the stroke.

The term Asymptote means a line which approaches nearer and nearer to some curve, but which, though infinitely extended, would never meet it. The clearance and vacuum lines of a diagram are asymptotes of a true expansion curve.

The letter B at the end of a diagram means that that end was taken from the bottom end of the cylinder.

A. B. or Aba. is understood to stand for above atmosphere, and B. A. or Bla. below atmosphere.

The term Compression is a term used to express the distance through which the piston moves in the cylinder after the exhaust has closed. Compression takes place between the piston and the cylinder-head at the end of each stroke; and the distance from the end of the cylinder at which it takes place depends on the amount of lap on the valve.

The term Cushion means the resistance offered on the opposite side of the piston induced by the steam shut up in the cylinder.

Cylinder efficiency.—This term is used to designate the amount of work performed in the cylinder of a steam-engine for a given pressure.

The term Clearance is used to express the extent of the space which exists between the piston, the cylinder-head, and the valve-face at each end of the stroke. See page 122.

Displacement. — This term is applied to the cubic contents, or the volume of water, steam, or air displaced by the piston during one stroke. It may be found by multiplying the area of the piston in inches by its stroke in inches. The product will be its displacement in cubic inches.

Duty. — This term is understood by engineers to mean the efficiency of steam-engines, or the number of pounds that an engine is capable of raising one foot high per second with an expenditure or consumption of one hundred pounds of coal.

The term Flexure means bending or curving. The point of flexure in a diagram is the point at which the cut-off closes and the expansion curve begins, as shown at *C*, explanatory diagram

No. 1, page 291. The point of contrary flexure is the point at which the line changes its direction by curving outwards and afterwards inwards, as shown at A, on diagram on page 291.

H. P. cyl. stands for high-pressure cylinder.

H. P. means horse-power, which, when applied to the steam-engine, means 33,000 lbs. raised one foot high; or 150 lbs. raised 220 feet high; or 550 lbs. raised one foot high in one second.

The term Hyperbola means a plane figure which is formed by cutting a portion from a cone by a plane, parallel to its axis or to any plane within the cone, which passes through the cone's vertex. The curve of the hyperbola is such, that the difference between the distances of any point in it from two given points is always equal to a given right line.

The term Isothermal means uniform or same temperature. As applied to an expansion curve, it means that such a curve represents correctly the expansion or compression of the steam when the temperature is uniform.

L. P. cyl. means low-pressure cylinder.

The term Ordinates means the vertical lines drawn across diagrams to facilitate the calculation of their power. See diagram on page 291.

The term Parallelism is generally employed, where two or more straight lines may be extended indefinitely, without any tendency to approach or diverge from one another. See atmospheric and vacuum lines on indicator diagrams.

Release. — This term is understood to mean exhaust. *Residuary* exhaust is that which follows the first release of the terminal pressure. The term *negative exhaust* is sometimes used, though not generally understood in its literal sense. It means compression or cushion, and absolutely amounts to the same thing, as it is

merely an early product of the exhaust, for the purpose of retaining a portion of steam in the cylinder as the crank approaches the centre of the stroke.

Rev. or Rev's is understood to mean revolutions per minute, though *rspm* is sometimes used.

I. H. P. means indicated horse-power. It means the number of H. P. of energy shown by the diagram of an engine, as found by multiplying together the area of the piston in square inches, its speed in feet per minute, and the mean effective pressure shown, and dividing the product by 33,000.

N. H. P. means nett horse-power, which is the I. H. P. minus the friction of the engine.

The term Initial pressure is generally understood to mean the pressure represented in the cylinder between the opening of the steam-valve and the closing of the cut-off. More properly speaking, it is the pressure represented in the cylinder at the commencement of the stroke, as the pressure frequently falls considerably before the closing of the cut-off.

M. E. P. means mean effective pressure. It is simply the amount by which the average impelling pressure exceeds the average resisting or counter-pressure. The M. E. P. on the piston of a steam-engine is the measure or exponent of the work performed.

The term Terminal pressure means the pressure at which the steam is exhausted from the cylinder, and may be said to be the exponent of the consumption of water by the engine.

The term Pipe diagram is applied to diagrams taken from the steam-pipe for the purpose of determining how much of the pressure of the steam in the pipe is lost in passing through the steam-ports to the cylinder.

The term Scale means the number of pounds of steam per square inch (acting on the piston of an engine) represented by each inch of vertical height on the diagram. Thus a 40 lb. scale means that each inch on the diagram represents 40 lbs. of steam per square inch, and so on.

The term Spring means the spring which is employed on the piston of the instrument, in order to resist the pressure of the steam and the vacuum. The following **table** will give the limit of pressure in the cylinder to which each spring may be subjected. The length of each spring given in the third column is such that each of them would be extended (when subjected to a perfect vacuum) to a length of $2\frac{7}{16}$ inches, which is the approximate length which would carry the pencil to the lower limit of the range of movement above given.

Scale of Spring.	Limit of Cylinder-Pressure above Atmosphere.	Length of Spring.
15 lbs. per in.	25 lbs.	2·192 ins. = nearly $2\frac{1}{5}$ ins.
20 " "	38 "	2·255 " = a little above $2\frac{1}{4}$ "
30 " "	64 "	2·315 " = " " $2\frac{3}{10}$ " or nearer $2\frac{5}{16}$ "
40 " "	90 "	2·345 " = nearly $2\frac{7}{20}$ "
60 " "	143 "	2·376 " = a little over $2\frac{3}{8}$ "
80 " "	195 "	2·391 " = a little above $2\frac{2}{5}$ "

To find the corresponding limit for grades not given, *multiply* the total range of movement, 2·625 inches, by the scale of the spring, and deduct the pressure of the atmosphere.

Example.— Suppose it is desired to find the limit of pressure for a 50 lb. spring: $50 \times 2{\cdot}625 - 14{\cdot}7 = 116{\cdot}55$.

The term String, as used in these pages, means the aggregate length of the ordinates of an indicator diagram.

The letter T on a diagram denotes that that end was taken from the top end of a cylinder.

The term Undulating means rising and falling, wavy. See dotted line on diagram No. 16, page 303.

Wire-drawing.—This term is applied to the common method of regulating the flow of steam from the boiler to the cylinder, by throttling or forcing the steam to ooze through some small or intricate device, such as the governor-valve, thus tending to destroy its elastic force.

The term Zero, when applied to indicator diagrams, means a vacuum.

How to Attach the Indicator.

Since it is of the first importance that the diagram should be correct, both as to its vertical and horizontal measurements, too much care cannot be taken in making the attachments. The best method of attaching the indicator to the engine is to drill and tap into the cylinder directly opposite the ports. When practicable, the holes should be located exactly at the centre of the clearance, or the space between the piston and the cylinder, when the crank is at the dead-centre; since, if the holes are bored in any part of the cylinder which is travelled over by the piston, the communication with the indicator will be closed at that point in each stroke. Care must also be taken that they are not too close to the cylinder-heads, as the projecting parts of the latter may interfere with the free flow of the steam between the cylinder and the indicator. But if such a difficulty should arise, recesses must be cut in the cylinder-heads, in order to establish the communication. If the heads can be removed for the purpose of locating the holes, it is always best to do so, as their exact location can be determined with perfect accuracy.

If circumstances will not admit of the holes being drilled into the clearance, they may be put in the heads; and, if it is not in-

tended to place the instrument in connection with both ends of the cylinder at the same time, this location is preferable. It is claimed by many engineers that reliable diagrams cannot be obtained, when the pressure has to be transmitted through a long pipe, as is the case when the instrument is connected to both ends of the cylinder. But it has been shown by experiment, that if the cylinder is tapped instead of the heads, thereby using the shortest pipe, and the stop-cocks are placed as near as possible to the instrument, the difference between diagrams so taken, and those taken from a direct attachment, is not always noticeable. If, instead of two stop-cocks, one on each side of the instrument, a threeway cock be placed under it, which will allow steam to be admitted from either end through the same plug, the difference can hardly be detected. If no such cock can be had, straight way-cocks of ample aperture, placed as close to the *L* or *T*, to which the instrument is attached, as possible, will give sufficiently satisfactory results for all ordinary purposes. When, however, it is decided to take the diagrams separately, two cocks become necessary, and the card must also have two loops or hooks, as far apart as the two positions which the instrument is to occupy. Then it may be quickly shifted from end to end as desired. If two instruments are attached to an engine, diagrams may be taken simultaneously from both ends; but, while such an arrangement obviates the difficulty of equalizing the events of the two ends with one instrument, it is open to the objection that, if there is any difference in the action of the two instruments, or in the strength of their springs, this circumstance will interfere with the comparison.

Motion of the Paper Drum.

Owing to the almost endless variety of engines, their peculiarities of design, etc., it is impossible to give very definite instructions which will be applicable to all cases. But it must be borne in mind, that the motion of the paper drum must be coincident with that of the piston in respect to its times of stopping and

starting, and be a miniature reproduction of it in all other respects; or, in other words, equal piston movements must be represented by equal movements of the paper throughout the whole stroke. To whatever the cord may be attached, whether to a temporary wooden pendulum fastened by a screw to a post, or to the beam of a beam-engine, it (the cord) must be at right angles to a line between its point of attachment and the pivot of the beam or pendulum to which it is attached, when the piston is in the middle of its stroke. For instance, suppose the engine to be horizontal, and that a wooden pendulum is attached to a light post set up by or on the engine, or some other convenient object, or is suspended from a joist, the lower end being connected to the cross-head, and that the point on the pendulum where the motion is sufficiently reduced for the paper drum is higher than the instrument, so that the cord must incline downward. In such a case, unless a carrying pulley is used to deflect it to a horizontal direction, or unless the point of attachment for the cord is moved as many degrees from the centre line of the pendulum as the cord inclines downwards, the movement of the pendulum will be too fast at one end of its travel, and too slow at the other, and the diagram will be distorted. The effects of such distortion will be to cause the ends to appear unequal when they are not so, or else to conceal or exaggerate inequalities where they really exist.

The length of the pendulum may be from one and a half times to twice the length of the stroke or more. If too short, the ends of the diagram will be distorted, unless the connection between it and the cross-head is sufficiently long. The pendulum may be attached to some object at the side of the engine, so that it may vibrate in a horizontal plane.

A good substitute for a pendulum consists of a drum about six inches in diameter, more or less, on the axis of which is another drum, the diameter of which requires to be as much less than the other as the movement of the paper is less than the travel of the piston. If the drum is mounted in a convenient position, and a cord from the large part is attached to the cross-head, and another

from the small part to the indicator, a spring in the large drum keeps its cord taut, just as the spring in the drum of the indicator keeps its cord taut.

When two instruments (right and left), are used, it is best to fix a sliding-bar alongside of them having the proper motion, carrying pins to which the cords are attached, these pins being placed between the two instruments so that the cord of each may pull towards the other, and the movement of the piston from either end of the cylinder may pull the cord of the instrument attached to that end, in which case the upper line of each diagram will be drawn while the cord is being pulled. But no perceptible advantage in the way of accuracy need be expected from this arrangement, though it is a very convenient one where a large number of diagrams are to be taken.

Analysis of diagrams.—All the various particulars which may be learned from the indicator diagram may be classed under three heads.

1. **Those** relating to the condition of the engine, such as its construction, adjustment, etc.

2. **The mean** or average pressure exerted on the piston as an element in calculating the indicated horse-power, I. H. P.

3. **The theoretical** rate of water consumption.

Here it is necessary to explain the terms used hereafter to designate the various parts of the diagram.

In diagram No. 1, *A A* shows the atmospheric line which is drawn when both sides of the piston of the indicator are exposed to the atmosphere. When tracing such a diagram it is preferable to pull the cord by hand, in order to make the atmospheric line longer than the diagram; *B C* is called the steam line. It is formed while steam is entering the cylinder. *C* is the point of cut-off. It cannot always be located exactly by inspection, as the closure of the part is generally sufficiently gradual to cause considerable fall of pressure before the port is entirely closed. In general, it may be located at the point where the outline of the figure ceases to be convex and commences to be concave. *C D*

is the expansion line or curve. *D* is the point of exhaust, which, like that of the cut-off, may be located at the point *of contrary flexure*, or that point where the expansion line begins to change the direction of its curvature. *D E* is the exhaust line. It commences at the point of exhaust, and may be considered as terminating at the end of the stroke, (though, strictly speaking, it does not terminate till the exhaust port closes at *F*.) *E F* is termed the counter- or back-pressure line, and by some the vacuum or exhaust line; but the former terms are more appropriate, as they are applicable to all diagrams, whether from condensing or non-condensing engines. In the diagrams of non-condensing engines it is above the atmospheric line, *A A*; while in condensing engines it is below; but in both cases it represents some counter-pressure, since a perfect vacuum is unattainable. *F* is the point of exhaust-closure. Its exact location cannot be so readily determined as the points *C* and *D*, as, although like the former, it is anticipated somewhat by a change of pressure, it is not marked by any change in the direction of the curvature of the line. In perfectly working engines it may be located geometrically, but it is seldom necessary to do so, since for all practical purposes it is sufficient to know where the change of pressure due to the closing of the exhaust begins, and its final result. *F G* is the compression curve, and *G B* is the admission line. These constitute all the lines which belong to the diagram proper, and all that are produced by the instrument.

For certain purposes the vacuum line *V V*, and the clearance line *H H*, diagram No. 1, are drawn, the former parallel to the atmospheric line, and at such a distance below it as will represent, according to the scale used, the pressure of the atmosphere as it was, or was supposed to be, at the time and place at which the diagram was taken. For this purpose it is usual, when great accuracy is desired, to consult a barometer at the time, and record its reading on the card; but, in the absence of a barometer, it is usual to assume the pressure at 14·7 lbs. per square

inch, which is the average at sea level; but, since the pressure diminishes at the rate of $\frac{1}{10}$ lb. for each 189 feet of elevation, allowance should be made for the known or estimated elevation of the locality. It should also be remembered that the pressure will vary nearly $\frac{1}{2}$ lb., and sometimes more, from changes in the weather.

The clearance line *H H*, **diagram No. I,** is drawn perpendicular to the atmospheric and vacuum lines, and at such a distance from the induction end of the diagram, that the space between them will bear the same proportion to the whole length of the latter as the whole volume of clearance bears to the piston displacement. When the amount of clearance is unknown, and it is not practicable either to calculate it or measure it by filling the space with water, it must be approximated as near as possible from the known clearance of engines of similar construction. The largest clearance will be generally found in the smaller sized engines of the ordinary single slide-valve type. Five such engines tested at the Cincinnati Industrial Exposition of 1875, had the following amounts 9, $9\frac{1}{2}$, 10, $11\frac{4}{5}$, and 12 per cent. of the cubic contents of the cylinder. Next to these will be the larger sizes of the same type, in which the clearance will range from 6 to 10 per cent. When two slide-valves are used with short, direct ports, but exhausting under the valves, the clearance will average from 3 to 6 per cent., according to the proportionate length of the stroke, the longest strokes having the smallest per cent. Corliss engines, in which the stroke is about three times the bore, have about 3 per cent. The least amount of clearance is obtained from valves designed to exhaust at both ends of the cylinder, instead of in the centre, as in the case of the ordinary single slide-valve. By such an arrangement of the steam- and exhaust-valves, the clearance has in many instances been reduced to $1\frac{1}{4}$ per cent. The clearance in *poppet-valve* engines is more difficult to calculate than in slide-valve engines; but, as a general thing, it does not exceed 5 per cent. It should be measured with water, when it is desirable to ascertain accurately the cubic contents of the clearance. *In poppet-valve*

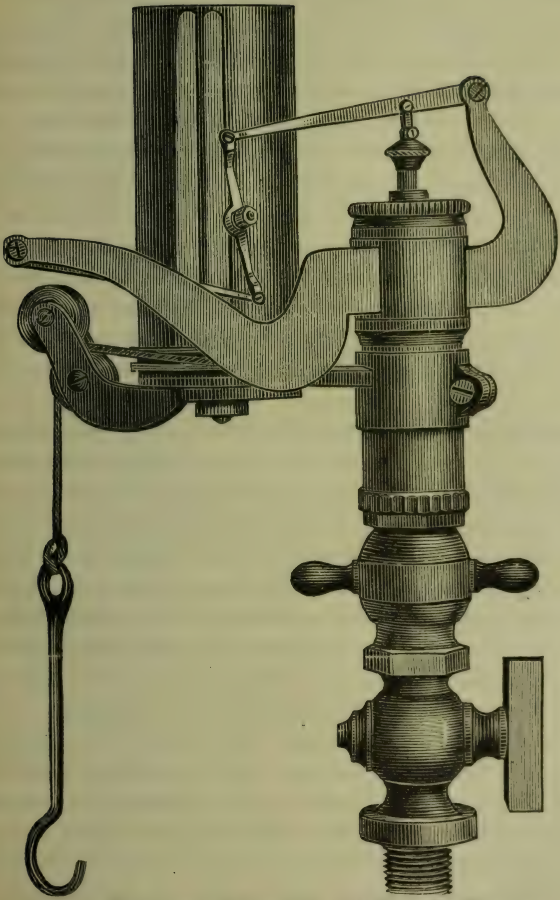
engines the cut-off and other events are independent and adjustable; consequently, diagrams taken from this class of engines are free from the limitations attending those taken from slide-valve, because advantage is frequently taken of their freedom of adjustment to give an earlier cut-off, or a later depression than is usually adopted with slide-valve engines. In all such cases the diagram will faithfully state the fact.

The Most Accurate Methods of Testing the Adjustments.

The conditions which are mainly instrumental in determining the conformation of the diagram are the valves and valve-gear, the length and capacity of the steam- and exhaust-pipes and ports, the design of the governor-valve, the condition of the valves and piston as to leakage, the amount of clearance, the speed of the piston, etc. The engineer may be called upon to analyze a diagram with reference to all the above conditions, or only to accidental derangements. In the first place, he must compare the diagram with one of the best form which can be produced in practice from the class of engines to which the one in question belongs; in the second case, he must discriminate between such defects as are due to accidental derangements and those that are due to design and construction, which cannot be remedied without the substitution of new parts. Suppose the engine to be of the throttling kind, of the best attainable construction and adjustment, its diagram should possess the following general features:

1. **The initial** pressure at *B*, diagram No. 1, should be as high at least as any subsequent pressure; and if the engine is not driving its maximum load, and the steam is in consequence more or less throttled, the pressure should begin to fall at *B*, and continue to do so at a tolerably uniform rate, until the point of cut-off, *C*, is reached.

2. **The cut-off,** when obtained by means of lap in the slide-valve, cannot, as a general rule, take place with advantage earlier



Richards' Parallel Motion Indicator.

in the stroke than about $\frac{5}{8}$, as the angular advance necessary to give any earlier cut-off would involve a too early exhaust and compression.

3. **From the cut-off,** *C*, to the release, *D*, is the expansion curve. Assuming the applicability of the Mariotte law to expanding steam, the shape of the expansion curve, *C E*, should be such that, if the distance from any point in it to the clearance line, *H H*, taken on a line parallel with the atmospheric line, be *multiplied* by the distance from the same point to the vacuum line measured vertically, the product will be the same for all points in the curve. Hence, if at the commencement of the curve, the two measurements are multiplied together, and the product divided by the distance from the clearance line to any other point, the quotient will be the distance of that point from the vacuum line, or the pressure at that point, if the pressure scale is used for the measurements.

4. **The release at** *D* will take place earlier or later, according to the amount of lap, both steam and exhaust, that is introduced. The steam-lap affects it indirectly, as, the greater it is, the greater the angular advance necessary to maintain the proper lead. Lap on the exhaust side affects it directly without change in the angular advance, by opening the port so much later, and consequently closing it so much earlier. The requirements of perfect working are, that it shall be early enough to release the piston of all undue back pressure before much of the return stroke is made, and late enough not to materially diminish the power of the engine. The conformation of diagram No. 1, page 291, shows about as early an exhaust as is admissible, because little or nothing would be gained by a later one, as the steam is not thoroughly exhausted until the piston has moved a short distance on its return stroke; and, while a later release would add a little to the average forward pressure, it would also increase the back pressure. Besides, a later release would involve either a later cut-off or an earlier compression; and, although the general practice is to place all these events later than is shown on the diagram, such practice is not calculated

to realize the best possible steam economy with that class of engines.

5. **The back-pressure line,** *E F*, should coincide with the atmosphere, or nearly so, in non-condensing engines and with the pressure shown by the vacuum-gauge in condensing engines. When it is in excess in either case, it indicates insufficient capacity in the exhaust-ports or pipes, or both.

6. **The compression curve,** *F G*, owes its form to the same laws that govern the expansion curve, and its degree of conformity to theory may be tested by the same methods. The only differences between the two are in the quantities of steam evolved in their production, and the order of their formation; the ending of one corresponding to the beginning of the other. As to the amount required to satisfy the best conditions, some difference of opinion exists. It is ascertained that a certain amount is advantageous, as a means of arresting the momentum of the reciprocating parts, while changing the direction of the force on the crank-pin in a more gentle and quiet manner, than would be done by the admission of steam as an opposing force. If the compression, or, more properly speaking, the cushion has fulfilled its functions, the induction will find the parts already prepared for the shock, and prevent a jar or thump. The maximum pressure reached by the cushion should never be greater than the average initial pressure; but within this limit considerable latitude exists, as, while it diminishes the power of the engine, it lessens the consumption of steam. The less the exhaust-lap, the earlier the exhaust will take place, and the later the compression, and *vice versâ*.

7. **The lead line,** *G B*, need not conform to any arbitrary standard. It satisfies the eye of the engineer best when it is vertical, or nearly so; but it may lean slightly inwards, indicating deficiency of lead, or outwards, indicating excess, without affecting the economy of the engine, and in most cases without sensibly affecting the smoothness of its running. In many cases the commencement of the lead line proper cannot be exactly located; but engines always run best when the compression and lead lines join

each other with an easy curve. When a single slide-valve is used, both the steam- and exhaust-lap must be provided for in its construction, and cannot be subsequently changed without a change of proportion. But since it is not the absolute amount of lap, but its amount relatively to the travel of the valve, which determines its influence, it follows that, by reducing the travel, the lap both steam and exhaust will be virtually increased, and *vice versâ.* Any change of travel must be accompanied by such change of angular advance as will maintain the proper lead. The adjustment of the cut-off by the link-motion of the locomotive is an instance of such change of travel and angular advance.

In the foregoing description all the capital letters refer to diagram No. 1, on page 291.

When two slide-valves are used, each performing the functions of induction and exhaust at its own end of the cylinder, the steam-lap may be increased by setting them farther apart, and diminished by contracting their connection; but in such cases steam-lap is obtained at the expense of the exhaust-lap, and *vice versâ.* Having learned from an engine embodying correct construction and performance the general features which should characterize a diagram, the *engineer* will have no difficulty in recognizing defects as well as deviations from diagram No. 1, on page 291. These conditions should be understood before the slide-valve, throttling engine diagram can be intelligently criticised.

Diagrams taken from Automatic Cut-Off Engines.

The points of difference between diagrams from automatic cut-off engines and those from slide-valve engines will be mainly found in the steam lines, the points of cut-off, and the expansion curve. When the automatic cut-off engine is worked in accordance with the theory of its operation, the steam is never throttled for the purpose of regulating the speed, but is admitted freely to the valves, the speed being regulated solely by means of variations in the point of cut-off. Hence, the steam line should indicate a

pressure equal to that in the boiler, whatever the load may be, and would undoubtedly do so, if the proportions were good and the valve-gear in perfect order. The necessary difference, then, between the throttling and automatic cut-off engine diagrams, may be thus stated. In the former, the height of the steam line varies with the load, the length remaining the same; in the latter, the length of the steam line varies with the load and pressure, the height remaining always approximately that of the boiler pressure.

The theoretical diagram. — From what has been said in the foregoing paragraphs, it is clear that a theoretical diagram may be constructed, representing perfect performance on the automatic cut-off principle, which cannot be done in the other case, as the height and conformation of the steam line depends on conditions too numerous and complex for analysis. Thus, with a given boiler-pressure for a steam line, a straight horizontal line may be drawn, corresponding with that pressure, and, from a given point of cut-off, an expansion curve may be drawn having the properties already described, and reaching to the end of the stroke. If the remaining terminal pressure is greater or less than the counter-pressure, a vertical line extending upwards or downwards to the height required by the counter-pressure will represent a perfect exhaust line. Then, for the return stroke, a line coincident with the atmosphere or a perfect vacuum, according as the engine is non-condensing or condensing, will represent the counter-pressure, and a vertical line up to the beginning of the steam line will represent the admission line and complete the figure.

If a compression curve is desired, it may be drawn through the assumed or actual point of exhaust-closure on the counter-pressure line, but such a curve cannot originate from a perfect vacuum. Hence, when the diagram is from a condensing engine, and the actual compression curve is to be tested by a theoretical one, the latter must be based on the actual counter-pressure present at the closure of the port.

This theoretical diagram being for the present assumed to be

perfect, not in the sense of representing the best conditions in an economical point of view, but only the most perfect performance possible under given conditions, is nevertheless the standard with which the actual one is to be compared, and by which it is to be judged. For this purpose it is customary to draw it around the actual, so that the imperfections of the latter may be readily seen and their magnitude estimated.

Application of the Theoretic Curve.

On tracing the theoretic curve on diagrams from different engines, a great difference in the degree of theoretical correctness shown in their expansion curves is revealed. The deviation from the theoretical is always in the direction of a higher terminal pressure, unless it is caused by excessive piston leakage. This may be explained on two suppositions, viz., leakage of the cut-off valves, and evaporation of the spray or water of supersaturation in the steam during expansion as the pressure decreases. Till recently the former was the only explanation offered, but, in more modern times, the latter has almost entirely displaced it. There is no doubt that both causes are in some degree responsible for the phenomenon, but the diagram itself seldom furnishes any reliable indications pointing to either cause to the exclusion of the other; nor does a study of the conditions under which the greatest incorrectness shows itself throw much light on the subject. As a general rule, large engines give more correct expansion curves than small ones, though numerous exceptions are met with in both cases.

Incorrectness is generally less with heavy loads than with light ones. But both the foregoing facts can be explained on either theory, since, with equal care in the fitting of the valves, a large engine will leak less in proportion to the amount of steam used than a small one. But the evaporation of the spray will be less perfect in the former than in the latter, owing to the longer time occupied in effecting a given degree of expansion, during which

the heat of the water, instead of being effective for evaporation, will be dissipated. In small, fast-running engines the steam undergoes a more rapid expansion, and the heat, rendered sensible by the removal of the pressure, has less time to be taken up by the cylinder walls, and is consequently more effective in vaporizing the moisture. Another fault in small engines is imperfection in the cut-off valves. Both causes afford better facilities to operate with an early than with a late cut-off, the longer time afforded by the former, and the less pressure under the valve, being favorable to the greatest leakage; and the greater the change of pressure is, the more favorable it will be to the evaporation of the moisture.

If, however, the deviation should (other things being equal) be found to be greatest when the water is high in the boilers, or when the steam is being rapidly generated, that fact would point to the spray theory as the undoubted cause of part of it. Such appears to be the case to some extent, though the observations taken on that point have not been numerous and careful enough to be of much value. But, whatever may be the cause of the phenomenon, it is so general that, whenever a very correct curve is met with, the suspicion of piston or other leakage, the tendency of which is to lower the pressure during expansion is always justly raised, and should be disposed of by test or otherwise, before such a curve can be confidently accepted as evidence of correct performance. Nevertheless, very correct curves are sometimes met with when piston leakage does not exist.

The most obvious lesson to be deduced from the facts at present in our possession, seems to be that, when any considerable incorrectness is met with in the curves of the diagrams taken from large engines, a considerable amount of leakage may be confidently inferred. But, in the case of small engines, particularly fast-running ones, the amount of incorrectness which may be caused by re-evaporation is undoubtedly greater than in large ones; but even in them the cut-off valves should not be too readily excused without examination.

The Initial Pressure, or Steam-Line.

A close approximation of the steam-line of an automatic cut-off diagram to the boiler pressure is rightly regarded as an indication of good construction and performance. Other things being equal, that engine which most nearly obtains the highest boiler pressure on its piston at the commencement of the stroke, may cut off the earliest — attain the highest ratio of expansion — and exhaust the steam at the lowest pressure. The last condition is the test of all improvements designed to promote steam economy, as, if they do not produce a lower terminal pressure for the same work, they do not fulfil the conditions for which they were intended. It is not sufficient to rely on the steam-gauge as a test of the steam-line, unless it has been recently tested and found correct. Most steam-gauges deteriorate by use, especially if exposed to undue heat or cold. When practicable, the engine should be stopped and blocked, or placed on the centre, steam admitted to the indicator at full pressure, and a line traced by hand. If this has been done, and a difference of four or five pounds between boiler and initial pressure be detected, how is the difference to be accounted for? The pipe may be too small, long, or crooked, or the ports be inadequate; or both these defects may exist. To test this matter, a connection should be made between the indicator and the steam-pipe above or below the throttle, as may be thought preferable. By means of this connection, a diagram representing the fluctuations of pressure in the pipe is produced over the engine diagram. A diagram produced in this way will show whether the loss of pressure is due to the ports or the pipe. In such a case, the pressure falls, when it is admitted to the cylinder, until it exceeds the initial pressure but little more than one pound. Then, as soon after the steam is cut off, as the space immediately above the cut-off valve can fill, the pressure rises, and the momentum of the steam in the pipe evidently carries it above that of the boiler, and about the middle of the stroke it falls again, evidently going below the boiler pressure. At about three-fourths of the

stroke it rises again, but this time not so high as it did at its first rise, probably not above boiler pressure. These secondary fluctuations possess no special significance, except as showing that the boiler pressure is to be determined by finding the mean of their extremes. Their frequency during the stroke will depend on the length of the pipe as determining their frequency in time, and on the speed of the engine as determining their relative frequency. The pressure of the steam also affects them, as high-pressure steam is denser than low. The trouble involved in making the necessary connection for such a diagram will of course exclude them in most cases, but their value to the engineer, as a means of arriving at correct proportions for the pipes and ports, will be apparent.

The Mean Effective Pressure.

Whatever uncertainty may attach to the inferences deduced from indicator diagrams, there is every reason to believe that, provided that the spring is correct, the instrument in good working order, and its indications mathematically calculated, the conclusions will be reliable. The usual method of calculating the mean effective pressure is to divide the diagram into any suitable number of equal spaces by lines or *ordinates*, to measure the centre of each space with the proper scale, and to take the average of the several pressures by dividing their sum by their number. But since it is easier to measure on a line than to guess at the centre between two lines, it will be preferable to make the first and last spaces half the width of the rest, which will make the lines stand in the centres of equal spaces. The measurements are then taken on them. Diagram No. 1, page 291, is lined in this manner. The most expeditious and accurate method of obtaining the average of these ordinates is to take a slip of paper, apply its edge to each of them in succession, and mark their combined length on it. This length in inches multiplied by the scale of the spring used, and the product divided by the number of ordinates measured, will give the desired average. By using a sharp-pointed

instrument, as the point of a knife, thrusting it into the paper at the foot of each ordinate, moving it to the top of the next, and carrying the strip with it, the measurement may be taken with great ease and rapidity.

The simplest method is to measure the ordinates between the direct and counter-pressure lines. This will give results accurate enough for most purposes; in fact, it will give the mean average of the two ends with entire accuracy, and this must always be obtained as a basis for calculating the power of the engine. But, since a diagram, from either end of the cylinder, represents, by its upper line, the pressure which impels the piston during one stroke, and by its lower, the counter-pressure which opposes it during the stroke in the opposite direction, it follows that, from either diagram alone, a corrrect balance for either of the two strokes which it represents cannot be struck. To do this, the mean counter-pressure of the one must be deducted from the mean impelling pressure of the other. To obtain these pressures separately, it is necessary to draw and measure the ordinates from the lines representing them to the vacuum line. This, however, is unnecessary, except for very accurate analysis. In general, the counter-pressure of the two strokes will be very nearly equal, especially if the exhaust and cushion are properly equalized; and even where they are unequal, the final average of the two ends will be correct.

The number of ordinates may preferably be one-fourth, one-third, one-half, or equal to the number representing the scale used; in which case it will only be necessary to multiply the combined length of the ordinates (or the *string*, as it may be called) by 4, 3, or 2, as the case may be, to obtain the desired result. If the number is equal to the scale, the multiplier, being 1, need not be used. Thus, suppose the scale to be 40, the number of ordinates 20, and the string $14\frac{1}{2}$ inches. As the scale represents twice the number of ordinates, the string being multiplied by 2 will give a product of 29 lbs. mean pressure. Or suppose diagrams to be taken from both ends of the cylinder, either on the same paper or separately, and the one to be calculated and averaged with the

other. The string of both may be taken together on the same strip, and if the scale is twice the number of ordinates, the length of the string will give the mean pressure at once without multiplying. When taking such a continuous string for two diagrams, the termination of the first should be marked with a pencil, so that the two may be compared.

When diagrams are met with, in which the expansion curve crosses the counter-pressure line, the string should be taken from the beginning up to the point where the lines cross, and after that in a reverse direction on the strip, so as to cancel the part of the string already made. When the terminal end is reached, what remains of the string first made will give the mean effective pressure (M. E. P.) in the usual way; or the total mean impelling and counter-pressure above vacuum may be found separately, and their difference ascertained.

To Space the Ordinates.

Draw vertical lines touching the ends of diagram No. 1, *A B E I*, and apply a rule across them in a more or less oblique direction, till some division on the rule, as $\frac{1}{16}$, $\frac{1}{12}$, $\frac{1}{10}$, $\frac{1}{8}$, or $\frac{1}{4}$, will divide the distance between the points where the rule crosses the lines, the desired number of two or three times the number of times. Thus the line *H C I*, in diagram No. 1, is $3\frac{3}{4}$ inches long, and contains the $\frac{1}{16}$ division 60 times; consequently, $\frac{3}{32}$ pointed off at each end, and $\frac{3}{16}$ for the other spaces, will correctly divide the diagram for 20 ordinates. With a little greater obliquity the distance would be 4 inches, when $\frac{1}{10}$ inches would be right for the end spaces, and $\frac{2}{10}$ for the rest.

To Calculate the Indicated Horse-Power (I. H. P.).

Multiply the speed of the piston in feet per minute by the area of the piston in square inches, and divide the product by 33,000. The result will be the H. P. for each pound of M. E. P., or the

H. P. value of each pound. See table on page 290. Then multiply the M. E. P. by this value. This method is preferable to multiplying by the M. E. P. before dividing, as, when several diagrams from the same engine representing varying loads are to be calculated, the value when once obtained will answer for all, the speed being practically the same in each case. The area of the piston-rod is generally ignored in such calculations, though it will diminish the area of one side of the piston about $\frac{1}{40}$.

Theoretical Economy.

If the steam used by an engine was known to be saturated, and at the same time free from any excess of water, and if it both entered and left the engine in that condition, it would be easy to calculate from the diagram the amount of water which the engine would use in a given time, supposing it to be practically free from leakage. Under such conditions the expansion and compression curves would conform rigidly to exact theory, and the total piston displacement for one stroke, divided by the volume of terminal pressure, and the displacement up to any point in the curve divided by the volume of the pressure at that point, would give the same result wherever the point was taken, which result would be the number of cubic inches of water used during that stroke. Unfortunately, the nature of steam is such that no exact calculations of water consumption can be made. Even if its exact condition as it enters the engine is known, as it may be by the calorimeter test, its capacity for receiving and parting with heat is so great that its condition changes immediately upon entering the cylinder, so that, after deducting the water of supersaturation, known to be present before it enters the cylinder, the diagram will still fail to account for all of the remainder. Nevertheless such calculations are frequently made, and as a means of ascertaining the relative economy of different engines, and of different loads, pressures, and adjustments in the same engine, they possess great value, since, whatever uncertainty may exist as to the unin-

dicated consumption, it may, so far as the engine is concerned, be assumed to be the same in each of the cases under comparison.

When it is desired to approximate as nearly as possible to the actual consumption by calculation, a certain amount must be added to the theoretical result. This amount varies from 10 to 50 per cent., according as the conditions are more or less favorable; but when they are so unfavorable as to require an addition of 50 per cent., they are obviously so bad as to call for repairs and changes, rather than elaborate calculations. When the conditions are generally good, a careful examination of them will make it possible to fix the margin of uncertainty within tolerably narrow limits. A large engine, with well-jacketed cylinder and tight-fitting valves and piston, will generally require at least 10 per cent. addition, independent of the percentage of unevaporated spray, which may exist in the steam with which it is supplied, and this, unless the boiler is so set as to superheat the steam, will require from 10 to 25 per cent. more. In fact, the margin of uncertainty due to the boiler is much greater than that due to the engine, as not only will differently constructed boilers vary greatly in the amount of unevaporated water given off, but great difference will be found to exist with the same boiler, according to the height the water is carried, the rapidity with which it is evaporated, the amount of impurities present in the feed-water, or which have accumulated in the boiler, and many other conditions. Thus a rapidly fired generator, containing a large area of heating surface in proportion to the amount of water and little steam room and superheating surface, may, and often will, give off nearly or quite as much unevaporated water as is contained in the steam. The only fair way to test the performance of an engine is to test the steam as it enters it, both as to moisture and heat. It should also be borne in mind that, according to Trowbridge's tables, the difference between the economy of engines of over ten cubic feet capacity of cylinder and those under one cubic foot, is about 12 per cent. in favor of the larger size.

How to Calculate Theoretical Rate of Water Consumption.

The total displacement per stroke in cubic inches divided by the volume of the steam at release pressure, and the quotient multiplied by the number of strokes per hour, will give the total cubic inches used per hour. This, divided by 27·648, the number of cubic inches of water per pound, will give the total number of pounds used per hour, which, if divided by the I. H. P., will give the result in pounds per I. H. P. per hour. This is the usual method; but, when the *rate only* is desired, a shorter process may be adopted, based on the fact that, from a given diagram, the result would be the same, whether the calculations are based on the actual size of the engine, or some other size is assumed, say a smaller size; as, although the total consumption would be changed, the divisor would also be proportionately changed.

Suppose the engine to be of such displacement as to develop one horse-power with one pound pressure, and that it is driven by that pressure of water instead of steam. It being but one horse-power, its total consumption per hour and per horse-power per hour will be the same. Being driven by water, its displacement will be its water consumption, which will be obtained as follows: A horse-power is 33,000 lbs. lifted one foot high per minute, or $33{,}000 \times 60 = 1{,}980{,}000$ lbs. per hour, or $1{,}980{,}000 \times 12 = 22{,}760{,}000$ lbs. lifted one inch per hour, which would be the displacement of such an engine in cubic inches, and consequently its consumption in cubic inches of water when driven by water. Then, taking 27·648 cubic inches of water per lb., we have $22{,}760{,}000 \div 27{\cdot}648 = 859{,}375$ as its rate of consumption in lbs. of water per H. P. per hour. Then, if the pressure were greater than one lb., the amount used would be as many times less than the above, as the pressure was greater than one lb.; and also, if it were driven by steam instead of by water, the amount used would be as much less, as the volume of steam at the terminal pressure was greater than an equal weight of water. It follows that if we divide

859,375 by the product of the mean effective pressure, and the volume of the total terminal pressure of the diagram under analysis, the quotient will be the desired rate, whatever the size and speed of the engine. The use of this constant number renders the operation more easy and short, and, except in the case of the compound engine, entirely independent of all data except those furnished by the diagram itself, the scale of indicator being known.

The terminal pressure for this and subsequent rules is found, when the exhaust takes place before the end of the stroke is reached, by continuing the expansion curve to the end of the stroke. In other words, it is not what the pressure may be at the moment of release, but what it would have been if it had not been released until the end of the stroke.

How to apply the rule to diagrams taken from compound engines when the strokes of the two cylinders are equal. Multiply the M. E. P. of the low-pressure cylinder diagram by the area of its piston, and divide the product by the area of the piston of the high-pressure cylinder. The quotient will be the pressure, which, acting on the low-pressure piston, will be equivalent in energy to that acting on the high-pressure piston. Then add this quotient to the M. E. P. of the high-pressure cylinder, and with its mean pressure so augmented treat it in all respects as an ordinary diagram. Or the process may be reversed, *i. e.*, the diagram from the low-pressure cylinder, with its M. E. P. augmented by the quotient of the product of the area and M. E. P. of the horse-power cylinder divided by the area of the low-pressure cylinder, may be treated as an ordinary diagram; but the result by this method will be less than by the first.

When the two cylinders have different strokes as well as different piston areas, multiply together the M. E. P. piston area, and stroke of the high-pressure cylinder, and divide the product by the product of the piston area of the low-pressure cylinder multiplied by its stroke. The quotient will be the amount to augment the M. E. P. of the horse-power cylinder before treating it as a simple diagram.

The same calculations may be more conveniently made by means of the following table; to use it, proceed according to the following rule:

Find under P the number which corresponds nearest to the terminal pressure of the diagram, and multiply the terminal pressure by the number opposite it to the right under W, and divide the product by the M. E. P.; the quotient will be the rate of water consumption in lbs. per 1 horse-power per hour.

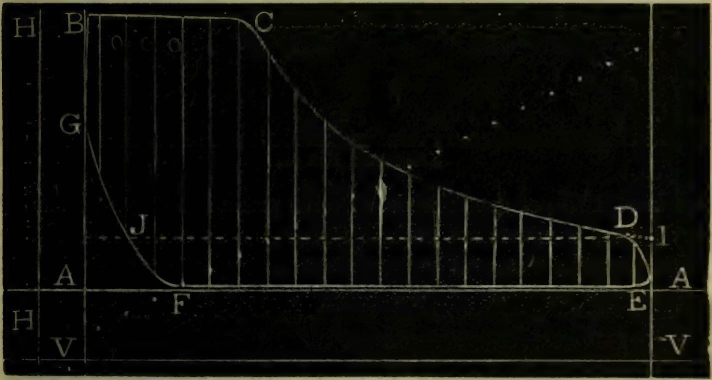
P.	W.	P.	W.	P.	W.	P.	W.	P.	W.
5	37·95	27	34·37	49	33·18	71	32·46	93	31·96
6	37·54	28	34·29	50	33·14	72	32·43	94	31·94
7	37·22	29	34·22	51	33·10	73	32·40	95	31·92
8	36·93	30	34·15	52	33·06	74	32·38	96	31·90
9	36·67	31	34·08	53	33·02	75	32·36	97	31·88
10	36·44	32	34·01	54	32·98	76	32·34	98	31·86
11	36·24	33	33·95	55	32·94	77	32·32	99	31·84
12	36·06	34	33·89	56	32·91	78	32·30	100	31·82
13	35·89	35	33·83	57	32·88	79	32·28	101	31·80
14	35·73	36	33·77	58	32·85	80	32·26	102	31·78
15	35·59	37	33·72	59	32·82	81	32·23	103	31·77
16	35·46	38	33·67	60	32·79	82	32·20	104	31·75
17	35·34	39	33·62	61	32·76	83	32·18	105	31·73
18	35·22	40	33·57	62	32·73	84	32·16	106	31·71
19	35·10	41	33·52	63	32·70	85	32·14	107	31·69
20	34·99	42	33·47	64	32·67	86	32·12	108	31·67
21	34·89	43	33·42	65	32·64	87	32·09	109	31·65
22	34·79	44	33·38	66	32·61	88	32·07	110	31·63
23	34·70	45	33·34	67	32·58	89	32·05	111	31·61
24	34·61	46	33·30	68	32·55	90	32·03	112	31·59
25	34·53	47	33·26	69	32·52	91	32·00	113	31·57
26	34·45	48	33·22	70	32·49	92	31·98	114	31·55

Example from same diagram. The terminal pressure is 25·5 lbs., and the mean of the numbers under *W*, opposite to 25 and 26 (34·50 and 34·41), is 34·45. The mean effective pressure being 30·5, the operation is as follows: $25·5 \times 34·45 \div 30·5 = 28·8$ lbs. per horse-power per hour.

As a matter of course, the theoretical rule of water consumption, as deduced from indicator diagrams, can never be fully realized in practice. It can only be approximated.

Indicator Diagrams.

All indicator diagrams are the perfect pictures of the performances of the engines from which they are taken, provided the indicator is in good order. There are two senses in which a diagram is said to be perfect or imperfect. First, it may be in perfect conformity to existing conditions, as clearance, load, steam-pressure, etc., though all of these conditions may be far from the best; or, second, it may not only conform to the above conditions, but it may represent the best attainable conditions, which would include no clearance at all, which is unattainable.



Explanatory Diagram No. 1.

In diagram No. 1, *B C* shows the steam line; *C*, point of cut-off; *C D*, expansion curve; *D*, exhaust; *D E*, exhaust line; *E F*, counter-pressure line; *F*, point of exhaust-closure; *F G*, compression curve; *G B*, admission line; *A A*, atmospheric line; *V V*, vacuum line; *H H*, line representing the clearance; 0 0 0, ordinates for ascertaining the average pressure; *I*, continuation of the expansion curve to end of stroke, to give the terminal-pressure for the purpose of calculating theoretical consumption; *J*, the point in the compression curve where the pressure equals the terminal; consequently, *I J* is the proportion of the whole stroke taken as the measure of the consumption.

Diagram No. 2 was taken from a Buckeye automatic cut-off engine 22 × 44; piston speed, 520 feet per minute; scale, 40;

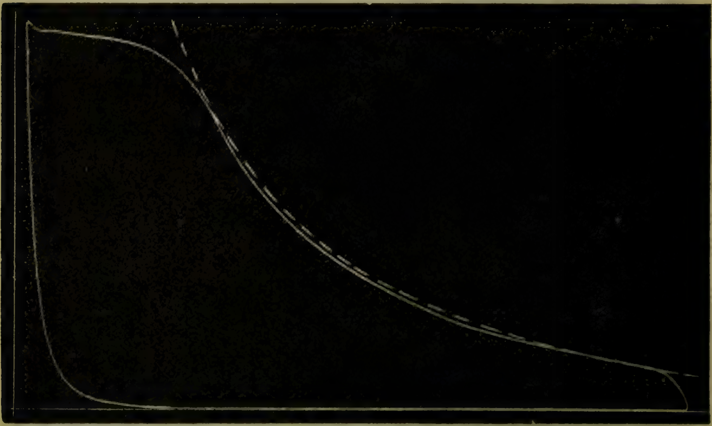


Diagram No. 2.

clearance, 1·75 per cent.; mean effective pressure, 36 lbs. It shows very perfect performance both of the engine and indicator.

Diagram No. 3 was taken from a locomotive built at the Baldwin

Diagram No. 3.

Locomotive Works, for the Pennsylvania Railroad Company, to run on the Philadelphia and Erie Railroad. Diameter of cylinder,

18 inches; stroke, 22 inches; speed, 93 revolutions per minute; boiler-pressure, 115 lbs. per square inch; initial-pressure, 100 lbs.; mean effective pressure, 86·60 lbs.; clearance, 4 per cent. At the time the diagram was taken, the engine was pushing a train of 15 loaded cars, whose gross weight was 302 tons, throttle-valve wide open, against a grade of 74 feet rise per mile. Adhesion per ton of load 600, resistance per ton due to grade 35·7 lbs. The slight rounding of the induction corner was probably caused by too much pressure on the pencil, which prevented it from rising till after the paper started to move. The diagram is very good. The expansion curve, as far as can be observed from its limited extent, is correct, and its compression curve very nearly so.

Diagram No. 4 was taken from a Wardwell valveless engine on exhibition at the Centennial Exposition held at Philadel-

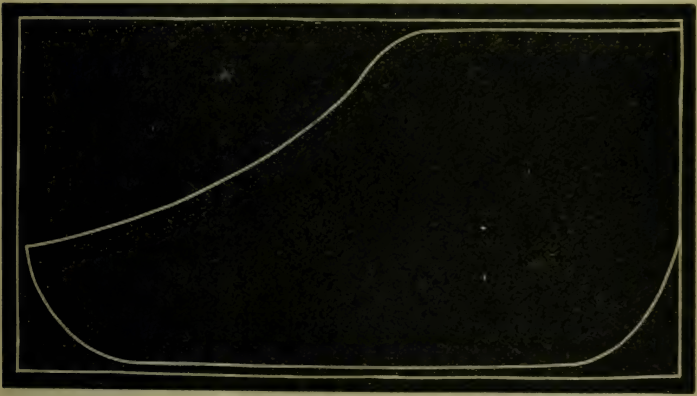
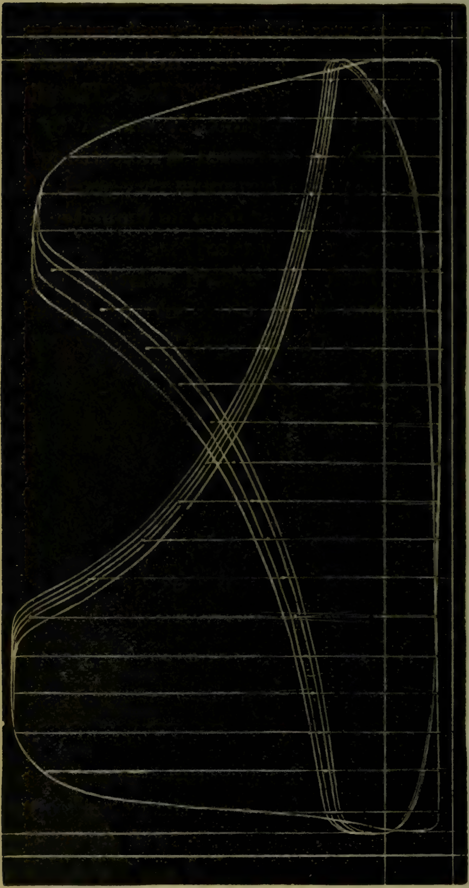


Diagram No. 4.

phia. The conditions under which the diagram was taken are not specified, but it will be observed that the exhaust-port opens quite late and quick, which explains the fact that the curve is all on the lower corner. The cut-off is quick and sharp. The induction and compression lines are also good. The lateness of the exhaust is a necessary result of the movement which produces it, as it is effected by a partial rotation of the piston-head, derived from

the lateral vibration of the connecting-rod, which gives a movement exactly equivalent to that of an eccentric without angular advance.

Diagrams No. 5 were taken from an old Corliss engine that had been running in the penitentiary at Jackson, Michigan, for about 25 years. Scale, 40; clearance about 3 per cent.; mean effective pressure, 47·5 lbs.; mean of the two ends, $47\frac{3}{8}$ lbs. It possesses no special interest, save to show the effects of adjustment due to long wear and use, without the application of an indicator or any other test. The excessively late induction would cause a perceptible loss of useful effect in the steam. The exhaust is much less perfect from one end than from the other, and much of the benefit of the vacuum is thereby lost. The pencil was held on during several revolutions, and, the governor being over-sensitive and fluctuating, different lines were drawn at each revolution.



Diagrams No. 5.

Diagram No. 6 was taken from a Harris Corliss engine operating at the Cincinnati Industrial Exposition of 1875. Size, 16 × 48; speed, 60 revolutions, or 480 feet of piston speed per minute. Both the isothermal, *I*, and the adiabetic curves are drawn. In tracing the latter, the following process was used. The horizontal lines, *A*, *B*, *C*, *D*, *E*, *F*, *G*, represent total pressures (above vacuum) of, respectively, 90, 80, 70, 60, 50, 40, and 30 lbs., the volumes of which are 298, 333, 378, 437, 518, 640, and 838. At the point, *H*, where the curve terminates, the total pressure is 19 lbs., the volume of which is 1290. Now, it is evident that if the distance, *H J*, which is 4·7 inches, represents 1290, the distance, *G J*, representing 838, (the volume of 30 lbs.,) will be proportionately as much shorter than *H J* as 838 is less than 1290. Hence, the formula, 1290 : 4·7 : : 838 : 3·05, or $\frac{4·7 \times 838}{1290} = 3·05$, will give this distance (3·05) from the clearance line, *J*, to

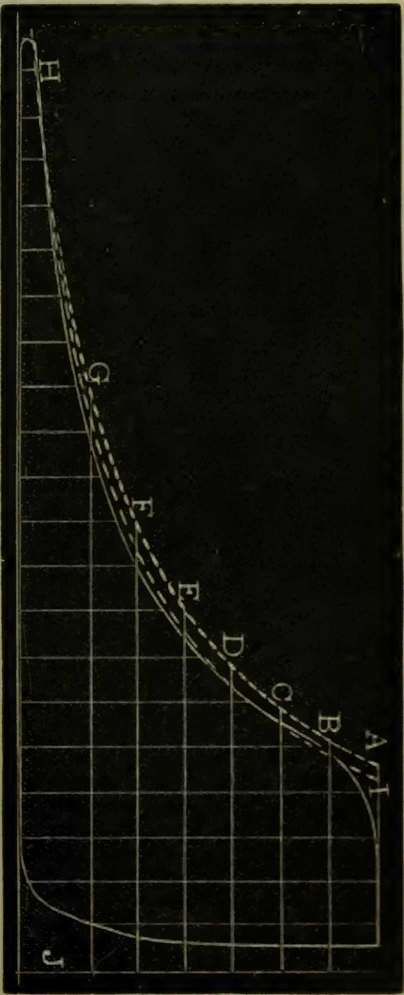


Diagram No. 6.

that point in the curve which shows a pressure of 30 lbs. In like manner the formula for the point, *F*, will be $\frac{4{\cdot}7 \times 640}{1290}$ for *E*, $\frac{4{\cdot}7 \times 518}{1290}$, and so on for the other lines, *D*, *C*, *B*, *A*. The foregoing process may, however, be shortened.

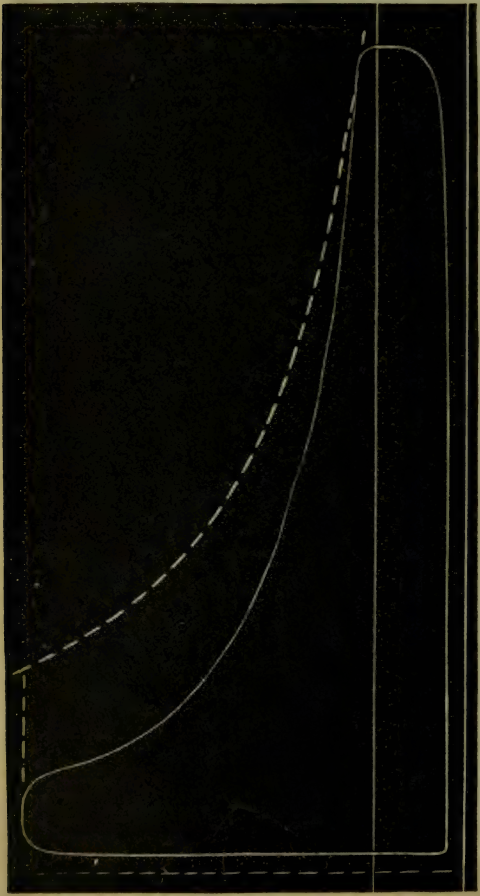


Diagram No. 7.

Diagram No. 7 was taken from a Holly engine located at the water-works of Rochester, N. Y. Size, 16 × 26·9 inches; speed not given, but it varies greatly, as it is regulated by the water-pressure; mean effective pressure, 30 lbs.; scale, 32 lbs. The cut-off valves of these engines consist of a single-poppet valve placed on the cover of the steam-chest, which cuts off the steam for both strokes; hence, all the steam in the chest is subject to expansion along with that in the cylinder, which has the effect of enormous clearance on the diagram. The theoretical curve

shown is not based on the actual clearance subject to expansion, but on a reasonably small amount, not greater than the average of true automatics of good construction; consequently, it is not a test of the conformity of the curve to the actual conditions, but rather a means of comparing the economical results of such an arrangement with engines of the best automatic type.

Diagram No. 8 was taken from a Wheelock automatic cut-off engine on exhibition at the Centennial Exposition. Size, 18 × 48; clearance, 4½ per cent.; scale, 30; mean effective pressure, 12; piston speed, 50 revolutions, or 400 feet per minute. *A* is the adiabatic and *I* the isothermal curve, both being based on actual terminal-pressure. The diagram is quite good for a light load, though the very slight compression is not in accordance with the weight of opinion as to what constitutes sound practice.

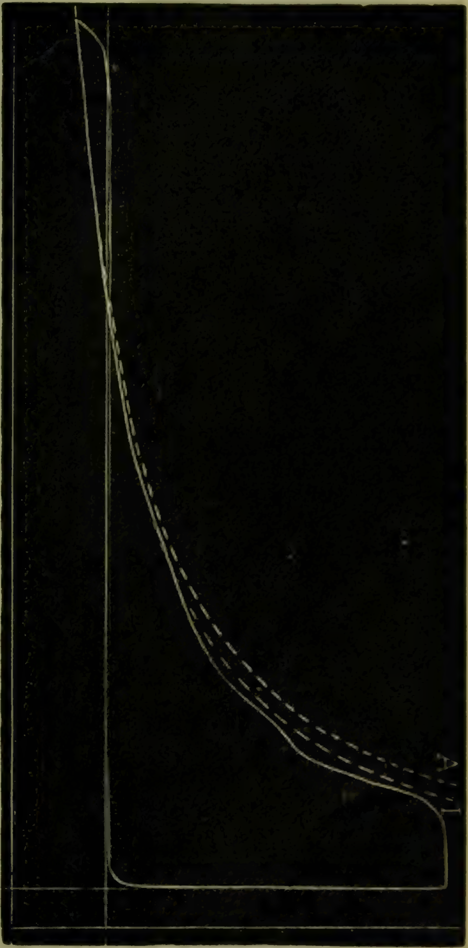
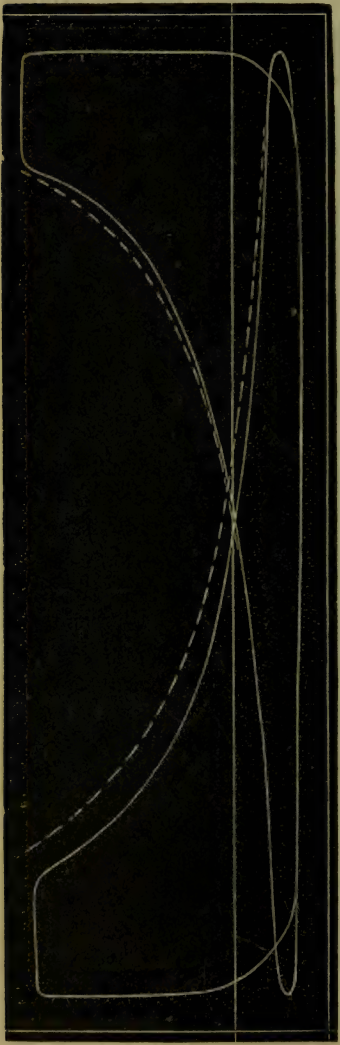


Diagram No. 8.

Diagrams No. 9 were taken from a **Cummer** slide-valve engine, with riding cut-off, built at Detroit, Michigan. Size, 26 × 36 inches; speed, 80 revolutions, or 480 feet per minute; scale, 30; mean effective pressure not given; clearance is unknown, but assuming it to be 4 per cent., which is about what its construction requires, the theoretical curve at one end shows correct performance, but that at the other shows considerable deviation. In such a case, taking the size of the engine into consideration, the explanation of this defect lies between two suppositions, 1st, that the cut-off valve leaked at one end and not at the other; or, 2d, that the volume of clearance is greater at one end than at the other. If the engine had been a small one, the supposition of the escape of the expanding steam from the right-hand end through a leaky slide-valve would be admissible; but the curve at that end is just what an engine of the size given should produce without leakage of any kind; hence, the left hand is the one to which attention is directed for the cause of the difference between the two, and the supposition of a leaky cut-off valve is the more probable one.



Diagrams No. 9.

Diagram No. 10 was taken from one of a pair of 16 × 30 inch single slide-valve engines, which were attached to the same shaft with cranks at right angles to each other. The piston speed was 350 feet per minute; mean effective pressure, 32·3. The sudden termination of the compression curve with a descending hook sug-

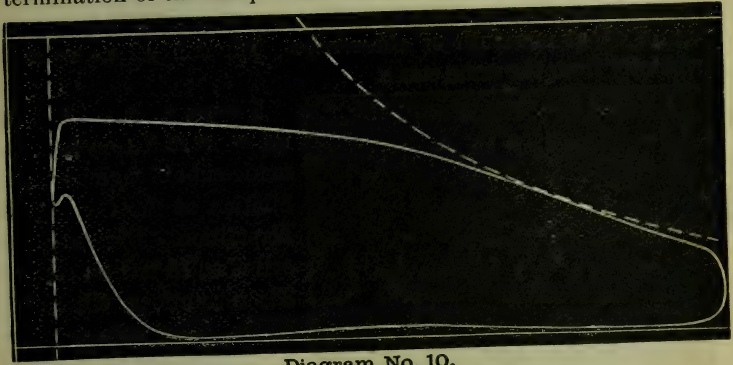


Diagram No. 10.

gests leakage of the piston or valve. The more rapid fall of the expansion curve than theory requires, strengthens this supposition, and points to the piston as the source of the trouble. The rise of counter-pressure in the middle of the return stroke is due to the reaction of the exhaust of the other engine.

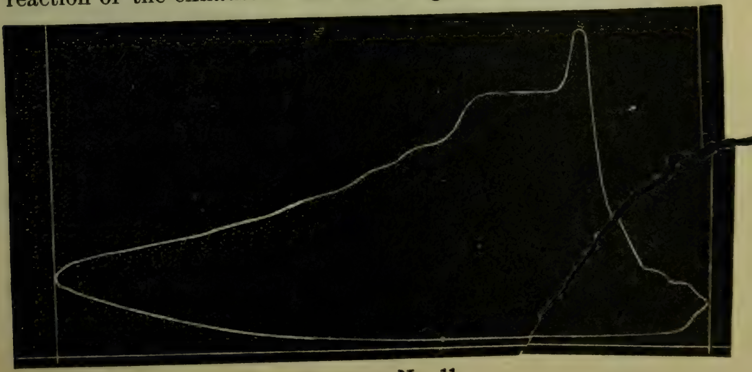
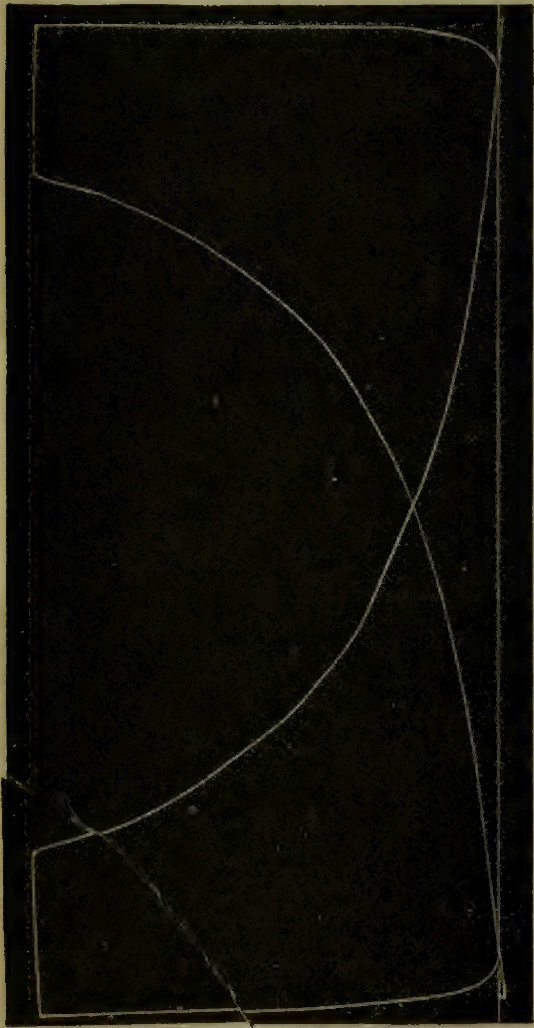


Diagram No. 11.

Diagram No. 11 was taken from an engine, 18 × 36, in a mill

in Detroit, Michigan. The cut-off was effected by a special cut-off valve above the steam-chest, operated by a Kendall's patent governor, which varies the throw and advance of an eccentric on the shaft by an arrangement similar to that of the link-motion of a locomotive. The most striking defect is the extremely late induction, showing a displacement of the eccentric, leading to a loss of about one-sixth of the stroke. The exhaust is too late, evidently from the same cause.


Diagrams No. 12.

Diagrams No. 12 were taken from a Brown automatic cut-off engine on exhibition at the Centennial Exposition. Diameter of cylinder, 15 inches; stroke, 38; revolutions, 65; scale, 30 lbs. They show wonderful conformity to theoretical requirements, and that the engine and indicator must be in the most perfect order to produce such cards. The unusually sharp cut-off corners are due to a certain extent to the fact that the induction and cut-off valves are of the gridiron type, and that the indicator is of an improved pattern, with exceptionally light moving parts; but nevertheless there is an air of suspicion about them, that will leave doubts of their genuineness in the minds of intelligent engineers who understand the action of the valves of steam-engines.

Diagram No. 13 was taken from a John Cooper engine, built under the Babcock and Wilcox patent, at Mount Vernon, Ohio. Diameter of cylinder, 20 inches; stroke, 36 inches; boiler-pressure, 55 lbs. per square inch; speed, 60 revolutions per minute; scale, 30 lbs. per square inch. It shows no imperfections worthy of note, except the imperfect retention of the compression-pressure, owing undoubtedly to leakage either of the piston or slide valve. Such a defect is a very common one, and may appear when no other evidences of leakage exist, in which case it is probable that, if the compression escapes by the piston, the leakage exists at the end of the stroke, or, if it escapes by the valve, only the portion which retains the compression-pressure fits imperfectly. In the present case the compression curve commences promptly, but succumbs completely, and falls again before admission, showing that the leakage commences suddenly near the end of the stroke.

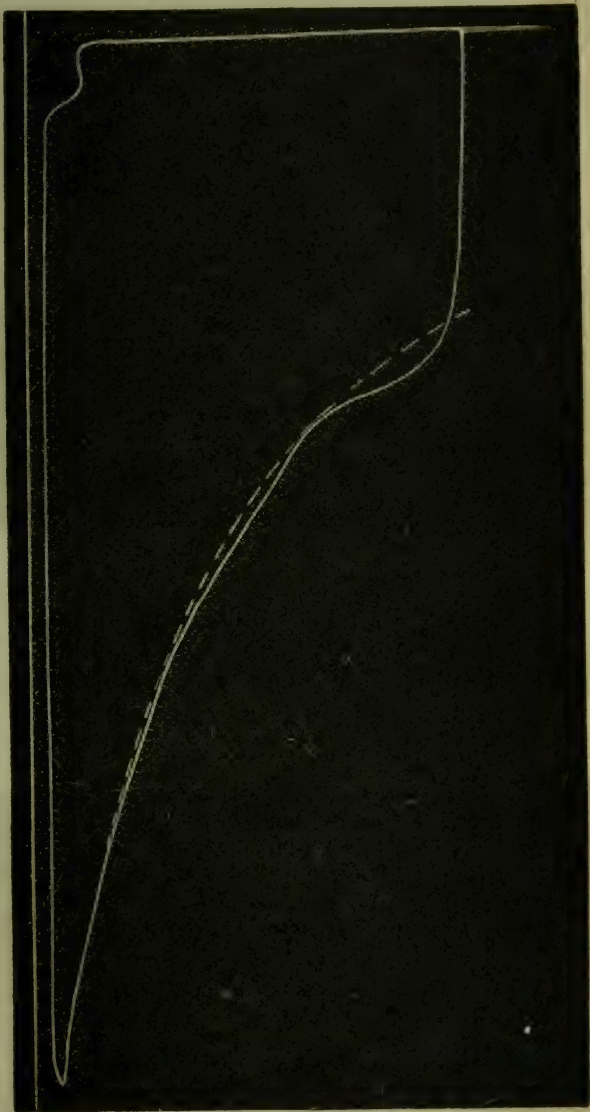


Diagram No. 13.

Diagram No. 14 was taken from a 9 × 15 high-pressure single slide-valve engine. Speed, 190 revolutions per minute; scale, 40;

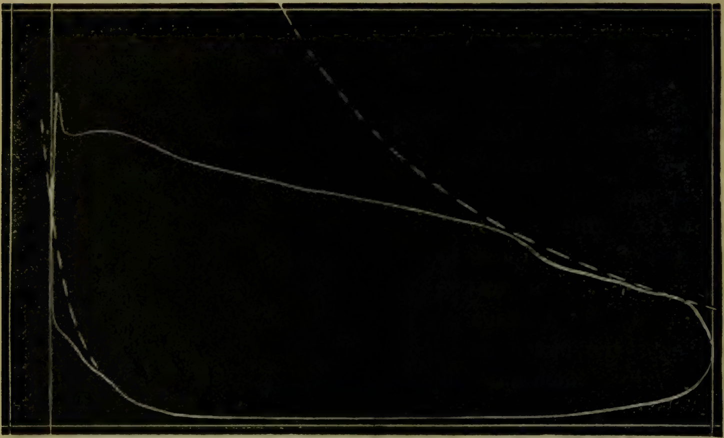


Diagram No. 14.

clearance, 6·4 per cent.; mean effective pressure, 41 lbs. It will be noticed that its events occur late, which defects arise from counter-pressure, indicating obstructed exhaust and imperfect rise in the compression-pressure, suggesting leakage of either the valve or piston by which the compression-pressure has escaped.

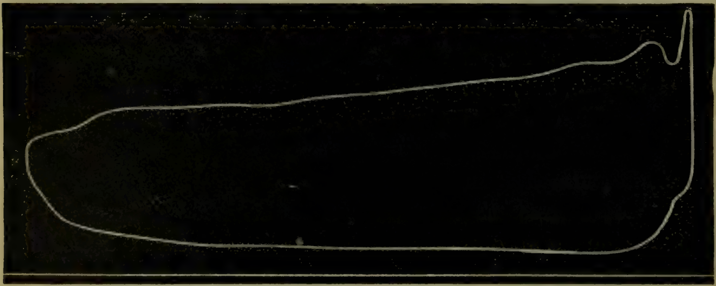


Diagram No. 15.

Diagram No. 15 was taken from the same engine. Its defective performance, as shown by its late cut-off, late and insufficient ex-

haust, and its excessive counter-pressure, all tending to extravagant fuel consumption, speak louder than words of the vital importance of an intelligent use of the indicator by engine builders, particularly when perfecting new designs and constructions. The counter-pressure was partly due to a contracted exhaust-nozzle used to create draught; but, even if it had had ample exhaust capacity at all points except at the nozzle, the counter-pressure created by that ought not to have exceeded one or one and a half to two pounds per square inch.

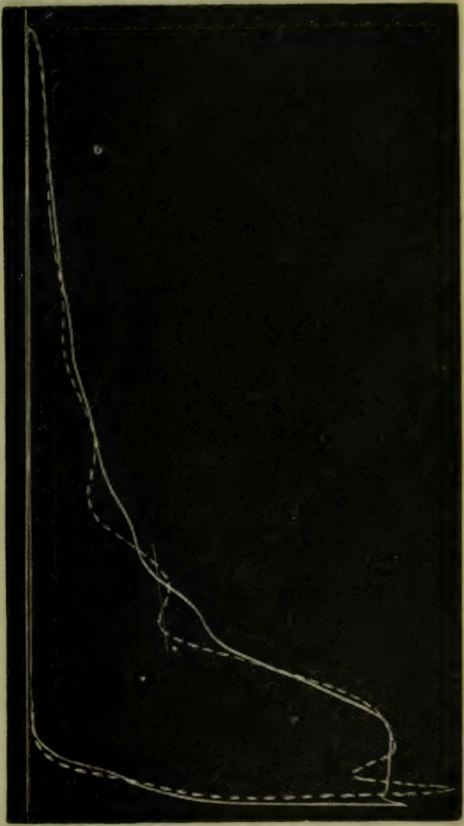


Diagram No. 16.

Diagram No. 16 is an exact transfer from two diagrams taken separately from the same end of the cylinder of an automatic cut-off engine. The dotted lines represent the card made by the Richards, while the plain lines represent that made by the Thompson, indicator. A comparison reveals the fact that the correct average pressure cannot be ascertained from a diagram which is distorted by vibration, and also that its indications are deceptive as to admission, cut-off, and compression.

Diagrams Nos. 17 and 18 were taken respectively from the high- and low-pressure cylinders of the compound engines of the steamship Pennsylvania, of the American Line, built by Cramp & Sons, marine engineers and naval architects of Philadelphia; speed, 58·3 revolutions per minute. The diagrams present no defects; the slight difference in the mean-pressure of the two ends of each card (as in the case of all vertical engines) is due to the unbalanced weights of the reciprocating parts.

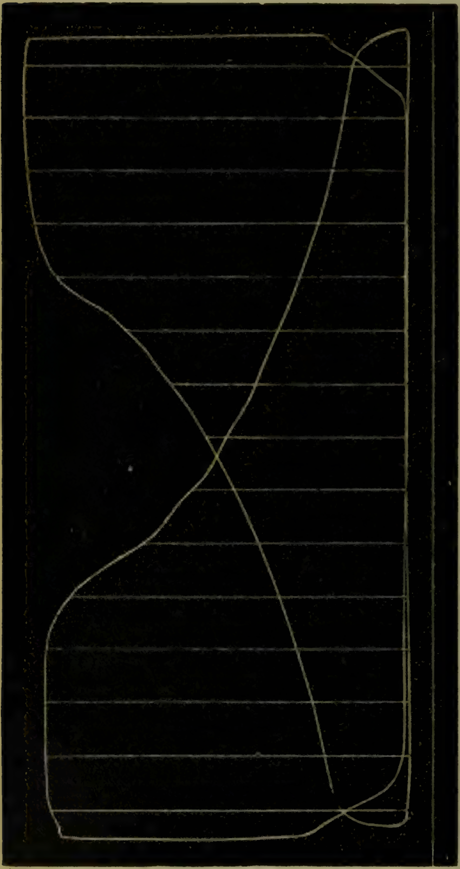


Diagram No. 17.

The theoretical clearance is about 10 per cent.; and, as this is probably not far from the actual, the expansion curves show very correct performance. The amount of vacuum shown is 10 to $10\frac{1}{2}$ lbs., which is above the average of marine engines.

As these engines are said to be more economical than any heretofore used on ocean steamers, a calculation of their theoretical economy will not be without interest. Taking the steam used

by the small cylinder as the measure of consumption, the first process is to find for it the equivalent of the mean-pressure acting on the large piston. The area of the small cylinder is 2574·1975 square inches, and that of the large one is 6379·4238 square inches. The M. E. P. of the small cylinder is 33·25 lbs., and that of the other 9·25 lbs. The rule is to multiply the area of the large piston by the mean-pressure acting on it, and divide the product by the area of the small piston. But, in the present case, it will involve less labor to perform the division first, that is, to divide the area of the large piston by that of the small one, and multiply the quotient by the M. E. P. of the large one. Thus, $6379·4238 \div 2574·1975 \times 9·25 = 19·33$ lbs., which, added to the M. E. P. of the small cylinder ($33·25 + 19·33 = 52·58$ lbs.), gives for it the equivalent of both, 52·58. Then the volume of the average terminal (28 lbs.) being 895, the calculation will be as follows: $\frac{859·375}{895 \times 52·58}$ $= 16.2$ lbs. From this the deduction for compression will be about 3 per cent., or ·48 lbs., leaving (16·2 — ·48) 15·70 lbs. per I. H. P. per hour, which justifies theoretically the claim made

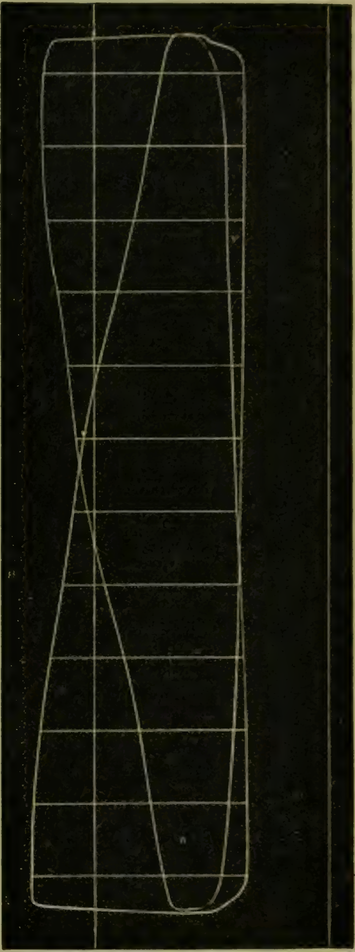


Diagram No. 18.

for these engines. The engines of the four steamships of this line gave very similar diagrams.

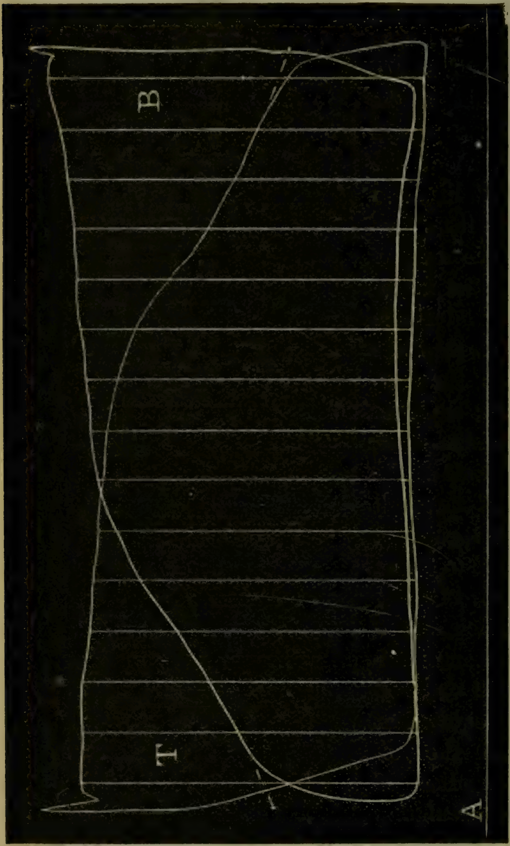


Diagram No. 19.

Diagrams Nos. 19 and 20 were taken respectively from the high- and low-pressure cylinders of the compound engines of the steamship St. Paul, built by Cramp & Sons, of Philadelphia, on her trial trip, and now plying between San Francisco, Cal., and Alaska. Scale of high-pressure cylinder 30 lbs., of low-pressure cylinder 12 lbs., per square inch. The data are as follows: steam, 67 lbs.; revolutions per min., 74; cut-off, ·25; vacuum, 26; indicated horse-power of high-pressure cylinder, 262·5; of low-pressure cylinder, 265·63; total, 528·13. Mean effective pressure of high-pressure cylinder, 43·125; of low-pressure cylinder, 14·25 lbs. The terminal-pressures, as shown by the diagrams, are as follows: The mean

terminal-pressure of both ends of the high-pressure cylinder is 47 lbs. (above vacuum); volume, 550. Of the low-pressure cylinder is 11·25 lbs. above vacuum; volume, 2100. The equivalents for each cylinder of the combined power of both are as follows: For the high-pressure cylinder, 43·125 + 43·64 = 86·765. For the low-pressure cylinder, 14·25 + 14·082 = 28·332. From these data, the calculation of the theoretical rates of water consumption will be for each cylinder as follows: For the high-pressure cylinder $\frac{859{\cdot}375}{86{\cdot}765 \times 550}$ = 18 lbs. per indicated horse-power per hour. For the low-pressure cylinder, $\frac{859{\cdot}375}{28{\cdot}332 \times 2100}$ = 14·44 lbs. indicated horse-power per hour.

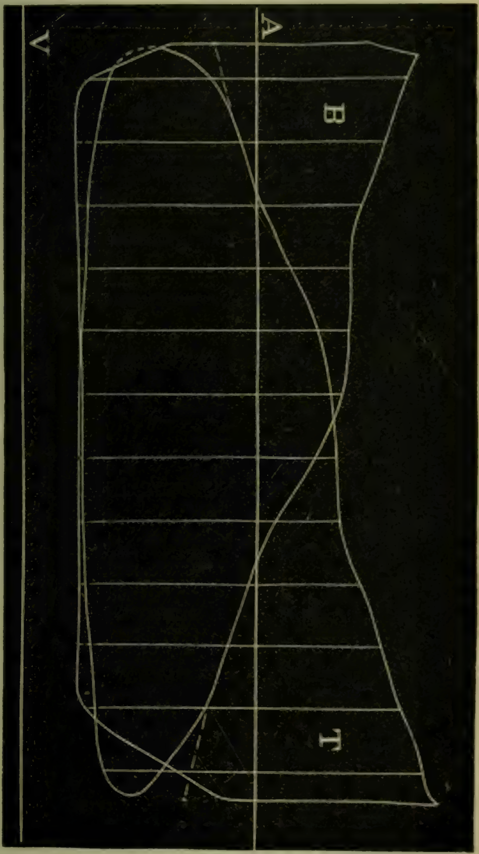
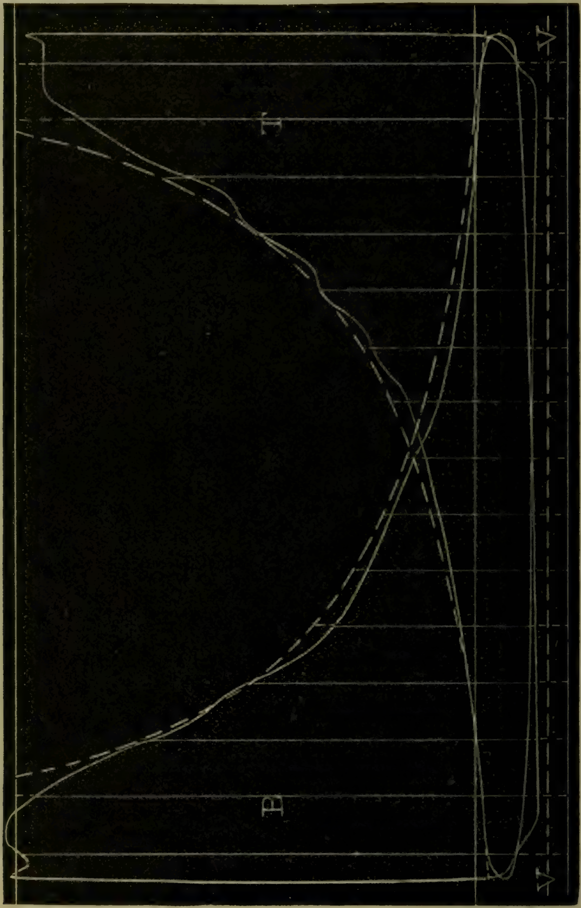


Diagram No. 20.

The maximum compression-pressures of each are so nearly equal to the terminal, that no correction for clearance and cushion need be made. The diagrams indicate good performance in all

respects, the lack of smoothness in the lines being presumably due to the tremulous motion of the vessel.



Diagrams No. 21.

Diagrams No. 21 were taken from the simple surface-condensing engine of the steamship Vera Cruz, of Alexander's Line, on her

thirty-ninth voyage from New York to Havana. Diameter of cylinder, $48\frac{1}{2}$ inches; stroke, 6 feet; speed, 60 revolutions, or 720 feet per minute; scale, 30 lbs. per inch. The boiler-pressure, which is represented by the lines above the diagram, was 72 lbs. per square inch; vacuum, 23 inches, the equivalent of which is represented by the dotted line *V V*. The full line below represents a perfect vacuum. The theoretical expansion curves are the adiabatic curves, calculated from the table of volumes on pages 39 and 43 of Roper's Hand-Book of Land and Marine Engines. The calculations are as follows:

Assuming the clearance to be 5 per cent., the mean-pressure of the theoretical diagram around the diagram *B*, which is based on the line *V V*, will be 32·8 lbs. The mean effective pressure of actual diagram *B*, 28·5 lbs. Percentage realized of the full theoretical value of the boiler, terminal, and condenser pressures, $\frac{28{\cdot}5 \times 100}{32{\cdot}8} = 86{\cdot}88$. Parallel calculations for the diagram *T* give the following:

Mean-pressure of theoretical diagram, . . . 31·8 lbs.
Mean effective pressure of actual diagram, . . 27·25 lbs.

$$\text{Percentage realized, } \frac{27{\cdot}25 \times 100}{31{\cdot}8} = 85{\cdot}68.$$

The mean of both ends is as follows:

Mean-pressure of theoretical diagrams, . . . 32·3 lbs.
Mean effective pressure of actual diagrams, . . 27·875 lbs.

$$\text{Percentage realized, } \frac{27{\cdot}875 \times 100}{32{\cdot}3} = 86{\cdot}3.$$

The area of the cylinder being 1847·45, and the piston speed 720 feet per minute, the horse-power value, or the horse-power for each pound of mean effective pressure, is calculated as follows: $\frac{1847{\cdot}45 \times 720}{33000} = 40{\cdot}3$. The mean effective pressure being 27,875 lbs., the total horse-power is $27{,}875 \times 40{\cdot}3 = 1123{\cdot}36$. The terminal-pressure is 13 lbs., the volume of which is 1842, and the

ıeoretical rate of water consumption will be found as follows:

$$\frac{859375}{1842 \times 27{\cdot}875} = 16{\cdot}74 \text{ lbs. per indicated horse-power per hour.}$$

The compression-pressure so nearly equals the terminal, that no correction for compression and clearance is necessary. The diagrams are in nearly all respects excellent; the curves, allowing for the unsteadiness which is apt to characterize diagrams taken from ocean-steamship engines, are remarkably correct; the engine was fitted with Corliss valves. The difference of about $2\frac{1}{2}$ lbs. between the vacuum attained in the condenser, *V V*, and that attained in the cylinder is a circumstance which is almost inseparable from such a high piston speed. A comparison of the rate of water consumption with that of such others as have been calculated, will be instructive, particularly with reference to the relative economic merits of simple and compound engines, a question which is yet unsettled. A comparison of the foregoing calculation with the ordinary or long process will be instructive, as showing the correctness of the short method and the vast amount of labor saved by it, especially when dealing with large engines.

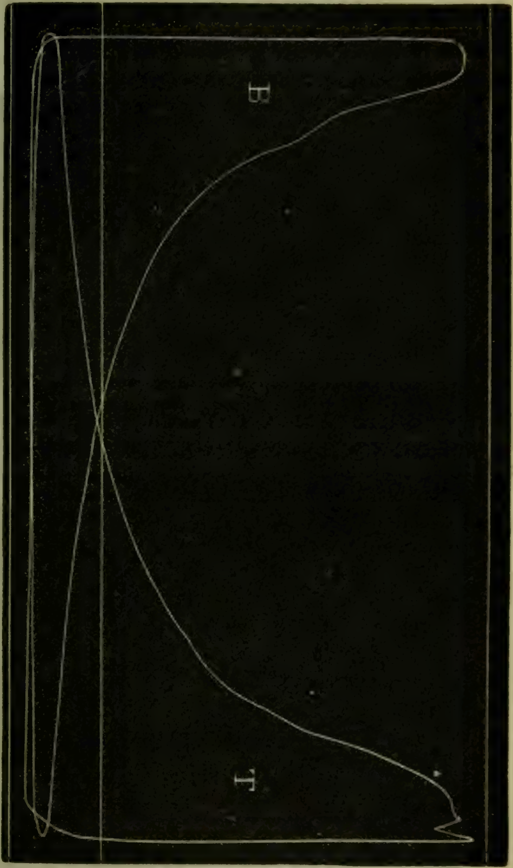
Thus, 720 feet per minute × 60 × 12 = 518,400 inches per hour, which, multiplied by 1847·45, (area of piston,) = 957,718,080 cubic inches per hour, as the displacement of the engine.

Then, 957,718,080 ÷ 27·648 (cubic inches of water per pound) ÷ 1842 (volume of 13 lbs. terminal) = 18,805·476 lbs. of water as its total theoretical consumption of water per hour; this ÷ 1123·36 (the indicated horse-power) = 16·74 lbs. per indicated horse-power per hour, as before.

In making a complete analysis of diagrams, a statement of the mean effective pressure, exclusive of vacuum and that due to the vacuum, ought to be given separately. Thus:

Mean-pressure, exclusive of vacuum, . . .	19·375 lbs.
Mean-pressure due to vacuum, ...	8·5 lbs.
Percentage of power due to vacuum, ...	30·5.

Diagrams No. 22 were taken from the same engine as diagram No. 21, on the steamer's forty-fourth return voyage to New York from Havana. It represents considerably lighter load than diagram No. 21, and shows the attainment of a better vacuum, is more perfect in its lines, and is equally correct in its expansion curves. The line above the diagrams represents the boiler-pressure. The calculations are as follows: Mean effective pressure of diagram *B*, 17 lbs. Mean effective pressure of diagram *T*, 19·5 lbs. Mean of the two, 18·25 lbs.



Diagrams No. 22.

Terminal-pressure of bottom diagram, . . .	6· lbs.
Terminal-pressure of top diagram, ...	7· lbs.
Mean of the two, ...	6·5 lbs.

Taking 3600 as approximately the volume of 6·5 lbs. pressure, the rate of water consumption will be 13·08 lbs. per *indicated*

horse-power per *hour*, which, if equalled, has never been exceeded by any other engines in this country, either simple or compound.

Diagrams Nos. 23 and 24 were taken from the simple surface-condensing engines of the steamship Knickerbocker, of Crom-

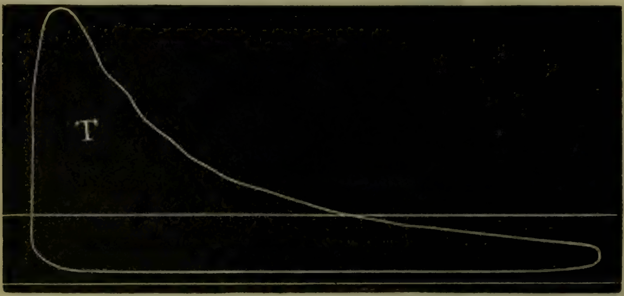


Diagram No. 23.

well's line, and running between New York and Boston. Many of the conditions could not be ascertained, but the mean effective

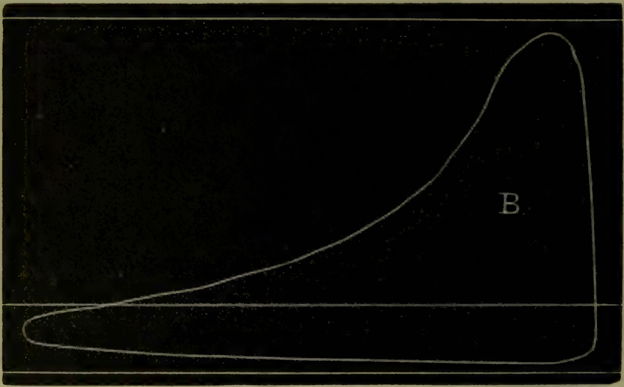
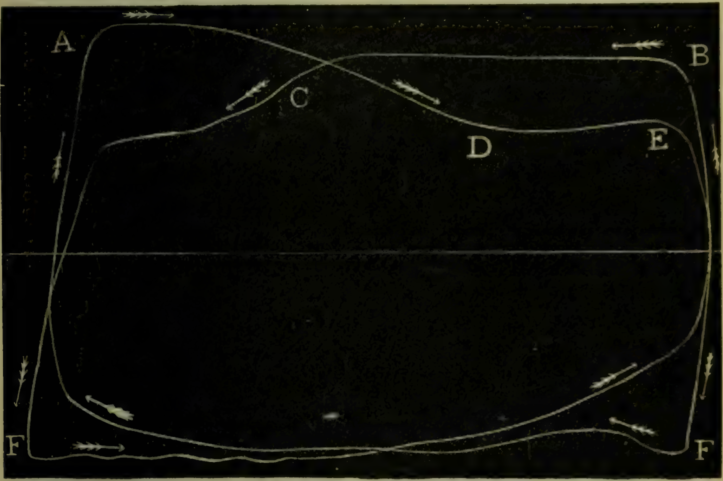


Diagram No. 24.

pressure of *B* appears to be about 29 lbs., and of *T*, 19 lbs. The calculations of the rate of water consumption give for the card, *B*, 13·74 lbs., and for *T*, 15·55. These very low rates are to some extent due to the very perfect vacuum attained. With the excep-

tion of the tardy induction, or deficient lead, as indicated by the inward inclination of the induction line, and the great difference in the work represented by the two, they are very perfect. And since both features may have been purposely introduced, the former to secure smooth running and the latter to compensate for unbalanced weight, etc., they should not be hastily pronounced faults of adjustment.

Diagrams No. 25 present a case of extremely difficult analysis, as none of the conditions under which they were taken could be



Diagrams No. 25.

ascertained. The left hand one shows tardy induction, by the inclination of the admission line to the right. From *A* to *D*, as will be observed, the pressure falls considerably; but it does not appear that the cut-off has taken place, as the curvature of the line is upward, which is never the case with a true expansion curve. From *D* to *E*, it will be seen, the pressure rises slightly, which renders it evident that the steam cannot have been cut off at any point previous to *E*, unless for an instant, after which it was readmitted. Supposing the line to correctly represent the

actual pressure on the piston, the most probable cause of the rise in the curve is, that the steam was admitted during the entire stroke to *E*, but not with sufficient freedom to maintain the pressure when the piston travel was greatest, or that the connecting-pipe between the cylinder and the indicator was long and tortuous. The right hand diagram is not so peculiar, as it shows a horizontal steam-line and a tolerably well defined point of cut-off, *C*, and expansion curve. In both the exhaust is much more free and prompt than the induction. The best vacuum was obtained at the beginning of the return stroke, *F F*, after which the lines undulate in a manner not easily accounted for, without an intimate knowledge of the construction of the engine and the conditions attending it.

Diagrams Nos. 26 and 27 were taken from the simple condensing engine of the steamboat Mary Powell, plying between New York city and Albany, which has exceeded in point of speed any other steam craft on American waters, or in Europe, so far as can be ascertained, making 25 miles per hour between those points with perfect ease.

Diameter of cylinder, ...	72 in.
Stroke of piston, ...	12 ft.
Diameter of piston-rod, ...	8 in.
Diameter of air-pump, ...	40 in.
Stroke of the air-pump, ...	62 in.

Very few data could be ascertained, but it seems that the M. E. P. of the top diagram was . . . . . 22·02 lbs.

Of the bottom, ...	22·23 "
Mean of both, ...	22·13 "
Terminal of top, ...	13·5 "
Of bottom, ...	18· "
Mean of both,.	15·75 "
Theoretical clearance of top, ...	12 per cent.
Theoretical clearance of bottom, . . .	17 " "

The water consumption appeared to be about 24·62 lbs. per horse-power per hour. The bottom card has the more compression. The size and speed of the engine could not be ascertained.

The Powell is a splendid specimen of the American beam-engine river-boat which some years ago were so great favorites on

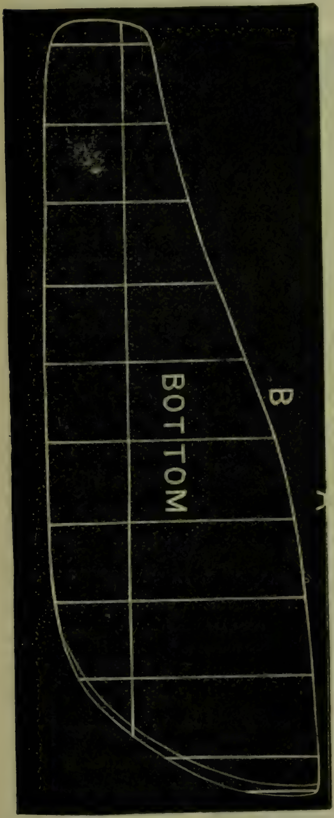


Diagram No. 26.

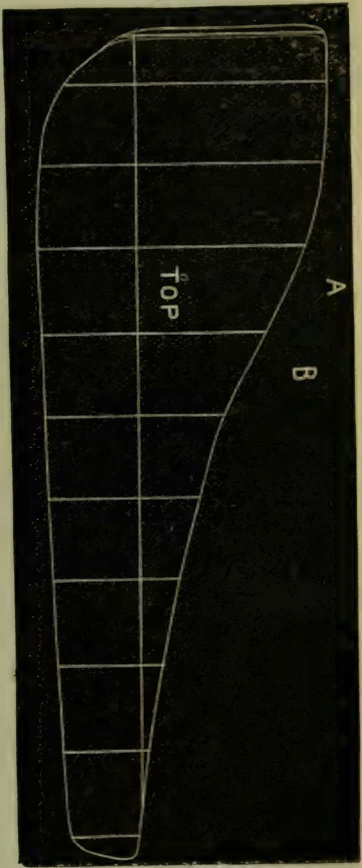


Diagram No. 27.

account of the great speed they were capable of developing, but which are fast disappearing, and being superseded by another class of engines, on account of inherent defects in their arrangement.

Formula for Finding the Theoretical Clearance when the Scale is known.

From two points in the expansion curve, as *A B*, the former as early and the latter as late as possible consistent with the certainty that both are in the expansion curve, draw the vertical lines, *A D* and *B C*, at right angles to the atmospheric and vacuum lines and the horizontal lines, *A C* and *B D*, forming the parallelogram, *A C D B*. Then, through *C D* draw a diagonal line, continuing it downwards till it intersects the vacuum line at *E*, and from this point draw a vertical line, which will represent the clearance. It will, in the majority of cases, indicate more clearance than actually exists; but if, as is sometimes the case with large engines of good construction and

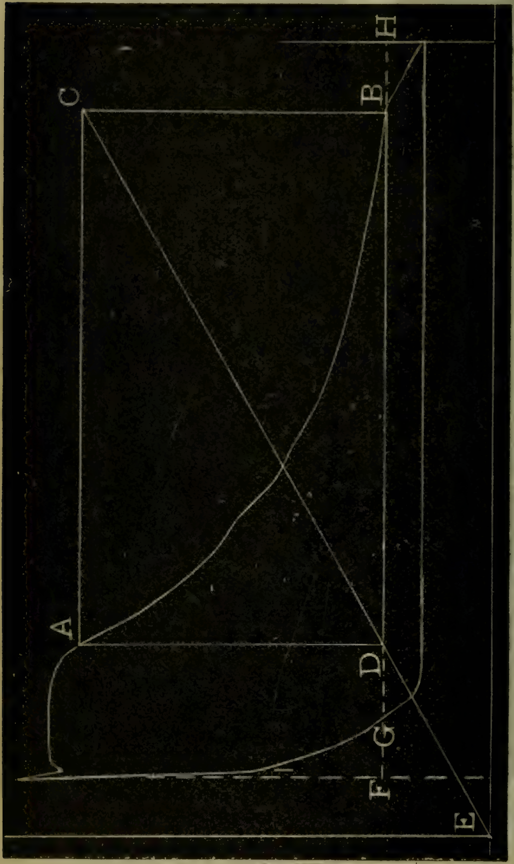


Diagram No. 28.

in good condition, the diagram agrees closely with exact theory, the clearance thus shown will be less than the actual.

On theoretical grounds, there should be no clearance at all, as any space between the cylinder-head and the piston at the end of the stroke must be filled with steam. But in practice it is impossible to dispense with it, since any wear of the parts must alter the stroke, and foreign substances, such as grease or water, may find their way into the cylinder. The loss resulting from clearance in cylinders may be lessened by judicious design, since, if compression takes place as the piston approaches the end of its stroke, it serves to raise the temperature of the steam enclosed, reduces the quantity of new steam required, and brings the momentum of the piston to rest, thereby lessening the shock on the crank.

Formula for Finding the Scale of a Diagram when the Clearance is known.

Draw a line representing the clearance; then proceed, as before, to draw the parallelogram, *A C D B*, and continue its diagonal, *C D*, till it intersects the clearance line, as at *E*. From the nearest point to this point of intersection, generally below, (which, by its distance from the atmospheric line, will represent the pressure of the atmosphere, according to one of the scales in use,) draw the vacuum line which fixes the scale. For instance, suppose the intersection occurs about $\frac{7}{16}$ of an inch below the atmospheric line. The nearest point below that point at which a vacuum line can be located to correspond with any of the usual scales is that corresponding with the 30 lbs. scale, or a little less than $\frac{1}{2}$ inch. If, however, there be reason to suspect that the actual scale varies from 30 to 40, (32, for instance,) this method will not determine it with certainty, but it will approximate it when the different scales used are known to differ from each other to the extent of 10 to 20 lbs. per inch. No method can be relied upon when only a limited length of the expansion curve is available, or when it is much distorted by vibration, or other defects in the performance of the instrument.

Formulæ for Finding the Horse-Power of Steam-Engines by Indicator Diagrams.

The custom of dividing the indicator card into ten ordinates has been generally adopted by engineers because ten is the most

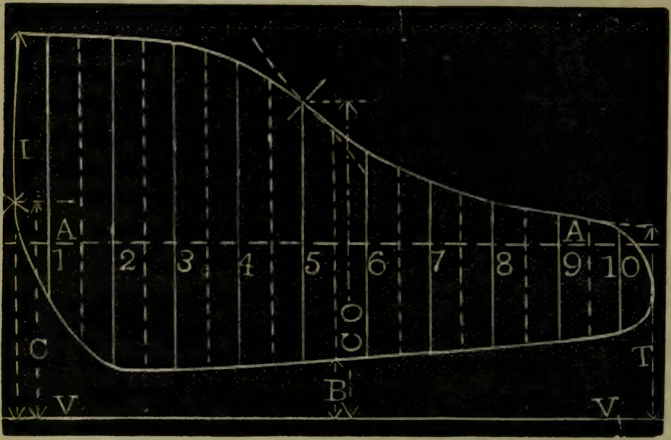


Diagram No. 29.

convenient number for a divisor, since the process of dividing by it consists merely of pointing off one decimal. The M. E. P. is ascertained by dividing the aggregate length of the ordinates by their number, and multiplying the quotient by the scale of the diagram. The following instructions will be found useful to persons unaccustomed to make the calculation.

First. — *Divide* the card into ten equal parts, as shown by the dotted lines in the above diagram, after which draw a line exactly through the centre of each space, as shown by the full lines 1, 2, 3, etc. Then draw the dotted line *A A*, representing the atmospheric line, also draw the full line *V V*, representing the zero, or vacuum line, which is equal to $14\frac{7}{10}$ pounds, below the atmospheric line; then measure the card at the following points:

The initial-pressure as shown at . . . . . *I.*
The pressure at the point of cut-off . . . . *C. O.*
The terminal-pressure at . . . . . . *T.*
The pressure at the end of the cushion . . . *C.*

Next measure the full lines, or ordinates 1, 2, 3, etc., with a slip of paper, marking with a sharp pencil or the point of a knife the length of each, until it contains the sum of all their lengths, which in this case will be found to be 11·75 inches; then, from the mean length $\frac{11{\cdot}75}{10} = 1{\cdot}175$ inches, and the mean-pressure $1{\cdot}175 \times 16$ scale of the indicator $= 18{\cdot}80$ pounds; the correct rendering of a card would be as follows:

Initial-pressure,	(above zero)	=	*I.*	=	32·0lbs.
Pressure at cut-off	" "	=	*C. O.*	=	28·0 "
Terminal-pressure	" "	=	*T.*	=	17·0 "
Mean back-pressure	" "	=	*B.*	=	5·6 "
Pressure at end of cushion	(above zero)	=	*C.*	=	18·5 "
Mean-pressure		=			18·8 "

Suppose the diagram to be taken from one end of a cylinder 50 inches in diameter (with a stroke of 48 inches), making 50 revolutions per minute, and the area of piston to be 1963·5 square inches, then $1963{\cdot}5 \times 18{\cdot}8 = 36{,}913{\cdot}8$. This pressure acts on the piston throughout the stroke, 48 inches, 50 times a minute, and the work done on one side of the piston in each minute would be $36{,}913{\cdot}8 \times 50 \times \frac{48}{12} = 7{,}382{,}760$. Now, if another diagram were taken from the other end of the cylinder, and the measurements be the same, the total work done by the engine each minute would be $3000 = 447$, indicated horse-power.

Another Formula.

In the analysis of diagrams in this work, the usual custom of dividing the diagram into ten ordinates has been departed from, because, in the first place, ten ordinates were not considered enough to insure accurate calculation; and, secondly, because, when the

number of ordinates is made the same, or one-half, one-third, or one-fourth as many as there are pounds per inch in the scale of the diagram, the calculation is, if anything, simpler than the old process, since the sum of the ordinates, as measured on the strip of paper in inches, is the mean effective pressure at once, when the number of ordinates equals the scale, and in other cases it bears the same relation to it that the number of ordinates does to the scale. Ten ordinates may be used, however, for such scales as are divisible by 10.

Suppose the scale to be 60, and the number of ordinates 10, and that the sum of their lengths is 7 inches. According to the former process, $\frac{7}{10} = \cdot 7 \times 60 = 42$ lbs.; by the latter method, supposing the number of ordinates to be $\frac{1}{6}$ of the scale, the process is simply $6 \times 7 = 42$; that is, the mean effective pressure would be six times the sum of the length of the ordinates, if the scale is six times their number.

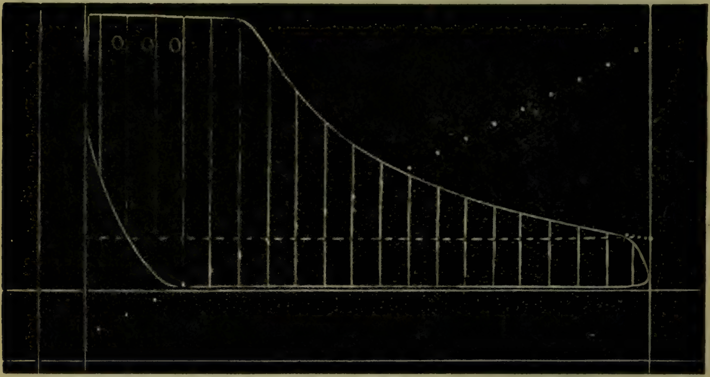


Diagram No. 30.

Suppose the scale to be 40 lbs per inch, one-half of that number, or 20 ordinates, as shown in the above diagram, are used; and suppose the sum of their lengths is found by the process of measurement above given to be 15·3 inches, then twice that number will be the mean effective pressure in pounds per square inch, or $15 \cdot 3 \times 2 = 30 \cdot 6$ lbs. Suppose the cylinder of an engine is 20

inches in diameter, 40 inch stroke, running at a speed of 75 revolutions, or 500 feet per minute; the area of such a piston would be 314·16 square inches; hence, $\frac{314{\cdot}16 \times 500}{33000} = 4{\cdot}727$ horse-power for each pound of mean effective pressure. The latter being 30·6, then $30{\cdot}6 \times 4{\cdot}727 = 145{,}656$, the indicated horse-power.

What Indicator Diagrams Show, and How they Show it.

The object of indicator diagrams is to show the pressure acting on the piston of the engine to which it is applied at all points, and also at what part of the stroke any change of pressure takes place.

Indicator diagrams supply the means by which to calculate the mean effective pressure acting on the piston, which, together with the known area and speed of the piston, furnishes the factors from which to calculate the power of engines.

Indicator diagrams show the steam-pressure by the height to which the pencil traces the line on the paper measured from the atmospheric or vacuum line.

When the line representing the back-pressure in the diagrams of high-pressure engines shows more than one pound above atmosphere, or, in low-pressure engines, two or three pounds more than the vacuum-gauge shows in the condenser, the diagram indicates undue back-pressure, and that there is evidently something wrong.

The diagram shows whether the valves of a steam-engine are properly set or not, because, if there is too little lead, it will lean towards the exhaust. If the exhaust takes place too early, the point, *D*, in diagram No. 1, page 291, will be further from the end, *I*; whereas, if the exhaust closes too early, and, as a consequence, there is too much "cushion" or "compression," it will be shown by the great distance of the point *F* from *E*.

A diagram shows whether the piston and valves are leaky or not; though it is often difficult to decide to which the leakage may be due, as the one neutralizes the other. But if the piston alone leaks, the effect will be a more rapid fall of the pressure during

expansion than theory requires, and the back-pressure will be greater than if the piston was tight. If the slide-valve leaks, the effect on the diagram will depend on the point at which the leakage occurs. It may leak at the ends, so as to keep on admitting steam after it covers the port; or it may leak at the bridges, and allow the steam to escape in advance of the exhaust. In the first case, the expansion line would fall less, and in the latter case more, than theory requires.

A diagram shows whether the steam is throttled or not by the expansion curve falling below the boiler-pressure when the throttle-valve is wide open.

A diagram shows the effect of small ports and small steam connections by the steam-line starting below boiler-pressure, and falling before the closing of the cut-off. A pipe-diagram is the only reliable means of determining such defects.

A diagram shows the effect of exhaust-lead, by the exhaust taking place before the end of the stroke is reached, as in nearly all the diagrams shown.

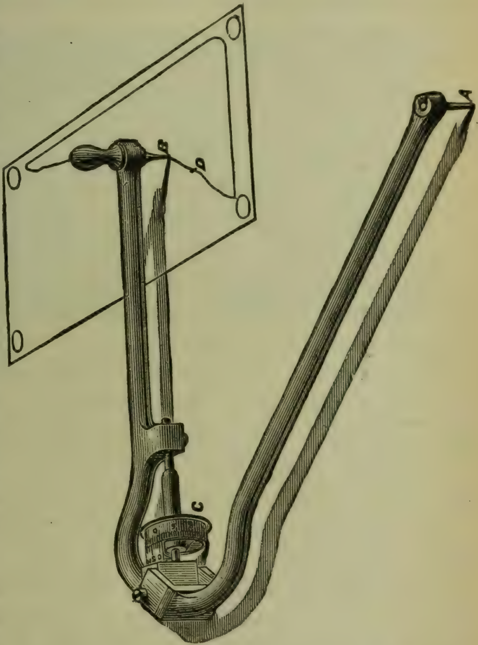
A diagram shows that the indicator is out of order, or whether there is lost motion between the piston and the pencil lever, by indicating more back-pressure than actually exists.

A diagram shows the point of cut-off, which may be termed the point of contrary flexure, that is, the point where the steam-line, *B C*, (explanatory diagram) changes its direction from a straight line to a curve.

A diagram shows the state of the vacuum in the condenser, and whether too much or too little injection-water is used or not; but in this case it is less reliable than the vacuum-gauge. Too much injection-water can only be shown on the diagrams, by taking one first with the proper quantity, and another with the increased quantity, and calculating the power of each. If the extra power, required to pump out the extra water against the atmospheric pressure, more than counterbalanced the gain from the better vacuum, the conclusion would be that too much injection-water was used.

The Planimeter.*

The planimeter, though not a recent invention, is almost unknown among engineers on this continent. This arises from the fact that, after its invention by Amsler, certain Swiss and German engineers got control of it, and limited the number that should be manufactured to their own individual necessities. It has never been manufactured in this country, or even offered for sale, until quite recently. Its functions are to measure indicator diagrams, irregular flues, steam passages, and all difficult or intricate figures. It gives at once the area of a figure, without any second measurement being required, as the reading shown on the index counter gives the accurate area in square inches of the diagram over which it had been passed.



To use the instrument, fasten the figure to be measured on a smooth board, and insert the point, *A*, in the board at any convenient location; then make a mark on the diagram, as at *D*; next fix the movable point, *B*, at the place selected for starting; then turn the index-roller, *C*, round until *O*, on its periphery, corresponds with the *O* on the fixed vernier; then move it round

* See page 656.

the figure to the right, or in the direction of the hands of a watch. After it passes round the entire figure, note how many whole numbers and subdivisions have passed the *O* on the vernier. The whole numbers will indicate the square inches, and the subdivisions tenths of square inches. If the *O* on the vernier falls between two subdivisions marked on the roller, read the number of square inches and tenths; then look on the vernier from *O* to 10, and find a mark which coincides with one on the rollers; the number of such mark, counting from *O*, will be the hundredths or second decimal place.

Thus suppose that, in the figure measured, six subdivisions and part of another one have passed, and that the fourth mark on the vernier coincides with a mark on the roller, the area of the figure will be either 3·64, 13·64, or 23·64 square inches, according to whether the roller has made less than one, more than one, and less than two, or between two and three revolutions. The eye can readily decide as to the number of revolutions the roller has made, as it would be impossible to make a mistake of ten square inches in estimating the area of a figure within the capacity of the instrument. If the figure measured is an indicator diagram, it will nearly always be of less area than ten square inches, or at most only a trifle more, as the utmost capacity of the indicator is $5\frac{1}{2}$ by $2\frac{3}{4}$ inches, or $15\frac{1}{8}$ square inches; and they are very seldom worked beyond 4 by $2\frac{1}{2}$ inches.

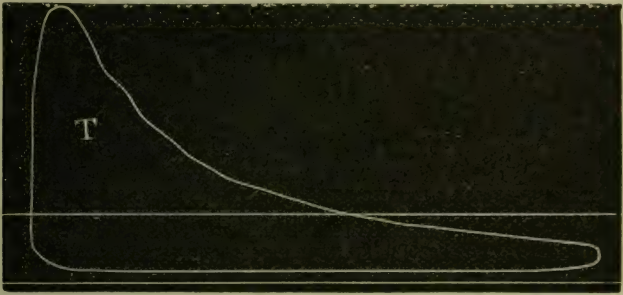
To find the mean effective pressure of a diagram from its area: Multiply the area by the scale, and divide the product by the length of the diagram in inches. Or divide the area by the length of the diagram, and multiply the quotient by the scale. The product is the mean effective pressure.

Example.—Suppose the area is found as above to be 3·64 square inches, the scale 40, and the length of the diagram is $3\frac{7}{8}$ (3·875) inches; $3{\cdot}64 \times 40 \div 3{\cdot}875 = 37{\cdot}65$ lbs., or $3{\cdot}64 \div 3{\cdot}875 \times 40 = 37{\cdot}65$ lbs.

It will be seen that the labor of calculation will be facilitated, if, in taking the diagrams, care is taken to make them even inches

in length. But as the engineer will have to measure many not of his own taking, he should have a rule divided into hundredths.

The annexed diagram was measured by the planimeter, and gives the following results: Area, 1·34 square inches; 1·34 multiplied by 40 the scale $\div$ 1·98 $=$ 18, the M. E. P.



The area of a figure may be taken without placing the *O* on the roller opposite the *O* on the vernier; but in such cases it is necessary to take the reading before and after the tracing is made; the difference between the two readings will be the area of the figure. But it is preferable to place the *O*'s together. The movable point of the instrument may also be turned to the left, but in this case the reading must be subtracted from 10 to give the true reading. Each of the figures stamped on the roller indicates a square inch of area, and if a figure contains 10 square inches at the tracing-point, the roller will revolve once, and the *O*'s will coincide as at the start.

Steam-Engine Economy.

Hardly a "decade" has passed since the days of Newcomen, which has not witnessed the promulgation of some vague scheme which it was claimed would revolutionize the economical working of the steam-engine, or even do away with it entirely, and super-

sede it by something else. Such wild schemes have invariably proved failures, as they must ever do, because there are some principles involved in the working of the steam-engine which, according to the natural order of things, can never be disproved. Consequently, those who intend to purchase steam-engines, or those who have capital invested in them, need entertain no fears that steam as a motive-power, and the steam-engine as a motor, will ever be superseded by anything else, while efficiency and economy are desirable objects to be attained.

Nor has there been any new principle discovered in connection with the steam-engine since "Newcomen's" time, as Watt, Hornblower, and Oliver Evans knew just as much about the latent and sensible heat, temperature, and the elastic force of steam as we do; though they lacked the knowledge of applying it so economically to the piston. This did not arise from ignorance of its properties so much as from the want of proper facilities to apply it. Nor is it at all likely that the steam-engine of the present day will ever be much improved upon in point of economy or efficiency, though it may be in point of durability. Good material, good tools, and perfect workmanship will go far towards the economical working of the steam-engine. It is a very noticeable fact, that no important improvement has been made in steam-engines of any kind within the past 15 years. To be sure, there have been many innovations introduced in that time, but upon examination it will be discovered that, in nearly all cases, they were a *revamp* of things which had been used before, and abandoned for want of experience in their use and proper facilities for perfecting them.

The mean effective pressure on the piston of a steam-engine is the exponent of the work performed. The term "effective pressure" means the amount by which the total pressure behind the piston exceeds that which acts on the other side in opposition to its movement. The *terminal*-pressure, or that at which the steam is released from the cylinder, is the corresponding exponent of the consumption of water by the engine or the cost of the power.

Hence, the best economy is attained when the *mean effective* pressure is highest relatively to the *terminal*-pressure, and anything that will increase the *former* without correspondingly increasing the *latter*, or which will diminish the latter without correspondingly diminishing the former, will improve the economy.

The amount of water consumed by an engine is the only intelligible criterion of the economical results it is capable of producing. The amount of fuel consumed will depend upon the kind of boiler used, its condition as to dirt, scale, etc., the manner in which it is set and fired, the quality of fuel used, the draught, and numerous other conditions; while the amount of water used will depend entirely on the engine, provided that it is furnished with dry steam. The theoretical rate of water consumption, as deduced from the diagrams, can never be realized in practice. A certain amount will always be lost from condensation, leakage, and unevaporated spray in the steam, for which no process of calculation can make allowance.

Now admitting that the evaporative efficiency of steam-boilers, under the best conditions, is 8 pounds of water per pound of coal, providing the water consumption of an inferior type of engine is one cubic foot, or $62\frac{1}{2}$ lbs., it would require $7\frac{3}{4}$ lbs. of coal, or its equivalent in other fuel, to develop a horse-power; while an automatic cut-off engine would yield a horse-power with a water consumption of 20 lbs., and the consumption of less than 3 lbs. of good coal. If an inferior type of engine require the consumption of 5 lbs. of coal per horse-power per hour, and an improved engine produce the same power from a consumption of 3 lbs., the latter will effect a saving of 40 per cent. in fuel over the former. Such comparisons may be considered extreme, but this is not the fact, as such cases are quite common in every manufacturing district. A manufacturer at Detroit, Michigan, was induced to take out an engine, which he was influenced to believe was wasteful, and replace it with one that was represented to be very powerful and economical, and at the same time very cheap. The engine was represented by the manufacturers as being capable of de-

veloping 100 horse-power; but it utterly failed to come up to this representation. When the indicator was applied, it showed that the engine was developing only 60 horse-power. The coal consumption was found to be nearly 8 pounds per horse-power per hour.

The great Lancashire (England) strike which occurred during the present year, and resulted in a loss to both employers and employees of several millions of pounds sterling, was brought about by an attempt on the part of the manufacturers to reduce the wages of the operatives one cent on every ten yards of manufactured cloth. They defended their action on the ground that ten per cent. was all the profit they realized on their manufactured goods, and stated that, unless the operatives would submit to the reduction, they would have to discontinue their business. Nevertheless, it had been well known for years, by reports made to the Lancashire Institute and the officers of the Midland Steam-Users Association, that there were thousands of steam-engines in that county, supplying power to factories, that were consuming from eight to nine, and in some cases ten and a half, pounds of coal per horse-power per hour, and yet the manufacturers could not discover any "*leak.*"

Before purchasing an engine or any other machine, there are some very important points to be considered which involve its commercial value, among which are, the amount which it would save or earn over another machine when in use, the time it would run without repairs, or the addition of any expenditure to its original cost. For these reasons, the conditions that should guide steam-users in the selection of engines are steady motion under varying circumstances, economy of fuel, and cost of maintenance. In the best types of the steam-engine, the principal expense, besides first cost, is fuel; but in inferior classes of engines, the cost of maintenance, such as lining up, and renewal of the different parts, increases annually, until in a few years the cost in many instances exceeds that of the fuel. It is to such considerations as these that steam-users should direct their attention when about to purchase steam-power, or replace a worn-out engine with a new one.

It has not always been the custom heretofore for those needing steam-power to purchase the most economical engines, but rather to buy for the lowest possible first cost, regardless of future maintenance. Manufacturers of inferior steam-engines being aware of this, agree to sell an engine of a certain horse-power for a certain price, perhaps 25 per cent. less than would be asked for a first-class machine.

Many persons are under the impression that it requires more fuel to carry steam at 100 lbs. per square inch than at 50 lbs., which is evidently a mistake, for while it requires a slight addition of heat to raise it from 50 to 100 lbs., the expenditure is more than compensated for by the superior expansion of the steam. Of course, the radiation will be greater at a 100 lbs. pressure per square inch than at 50 lbs.; but this would be more than overbalanced by the saving in the consumption of steam; as steam at 70 lbs. pressure per square inch will perform more than seven times as much duty as steam at 25 lbs. pressure.

Another fact not generally as well known to engineers and steam-users as it ought to be, and which illustrates the benefits to be derived from expansion, is, that if an engine was taking steam whole stroke, or $\frac{17}{18}$ of the piston-stroke, with say 60 lbs. pressure per square inch, if the pressure is raised to 75 lbs. per square inch and cut off at $\frac{5}{8}$ stroke, the engine would do the same amount of work.

Location of Steam-Engines.

There is no class of machines, save, perhaps, steam-boilers, that are so often injudiciously located as steam-engines; they are not unfrequently stowed away in out of the way places, without any regard being paid to their general appearance. This arises from the fact that persons consulted on such matters are allowed to locate steam-engines who are totally unfit to do so, on account of a want of that practical skill and experience that should be possessed by persons who undertake this duty.

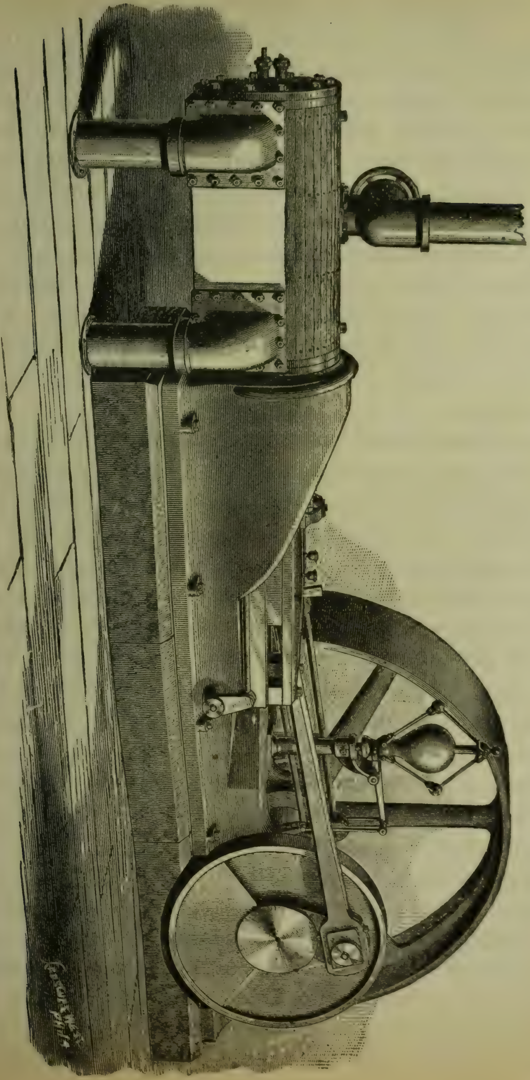
It is a mistake to locate an engine at the extreme end of a

building or a long line of shafting, as, if the engine were located to divide the work equally, the strain on the section carrying the driving-pulley will be only one-half what it would be subjected to if the motion were communicated from one end. Engines are frequently so located for the ostensible purpose of economizing space; but, on examination of the surroundings, it will, in a majority of cases, be found that no more space need be occupied by placing it in the centre, while there would be unquestionably a gain in the diminished friction.

The Porter-Allen High-Speed Engine.

The cut on page 331 represents a front view of the Porter-Allen High-Speed Engine. As will be observed, the bed-plate is of the box form, the surface of which is raised near the centre line. The main-bearing being brought as near as possible to the centre line of the cylinder, the breadth and depth being of sufficient proportion to resist all strains, it presents no sharp angles that would be a source of weakness. In fact, the bed-plate was designed to secure absolute rigidity under high-piston velocity, and correspondingly high steam-pressures, as well as a support for the cylinder, thus making a self-contained horizontal engine. The cylinder is overhung from the front end of the bed-plate, without any support excepting that which is derived from the butt-joint which it forms with the housing, which admits of equal expansion on all sides, and prevents the possibility of getting out of line. The rigidity of these engines may be judged from the fact that not the slightest vibration is experienced under the highest piston-velocities and steam-pressures; nevertheless, when of necessity they have long stroke, there is a temporary support placed under the cylinder.

The steam- and exhaust-chests are located on opposite sides of the cylinder, and cast in one piece with it; and the valves and seats are so arranged, that the removal of a small bonnet gives easy access to them. At the back of the steam-valves, an adjust-



The Porter-Allen High-Speed Engine.

able pressure-plate is introduced; this plate is made of a form which is calculated to resist the steam-pressure without deflection, and which is held against two inclined supports above and below the valve by a bolt screwed through the bottom of the chest. If this bolt is backed sufficiently, the steam will cause the pressure-plate to seize the valve. By turning the bolt forward, the pressure-plate is raised, and also moved away from the valve. The engineer can test each valve for leakage, by unhooking the disengaging hook, with which all these engines are provided, and work the valves by hand under steam-pressure by the starting-bar, while the assistant adjusts the bolt on the pressure-plate. If there has been no wear, the slightest movement of the bolt will cause the valves to be seized; but if there is any wear, it can be adjusted in a few moments.

The exhaust-valves work under the pressure in the cylinder, and have short movements, each valve opening four passages for the release of the steam. The exhaust-valve seats are formed on the covers of the chambers in which they work, and on which, also, the outlets are cast. They drain the bottom of the cylinder; they are very conveniently arranged for facing, adjustment, or repairs, and, by removing small bonnets, they may be taken out and replaced without any inconvenience. The release of the steam is one of the most remarkable features of these engines, as the link gives to the valve an admirable movement in every respect. The movement of the exhaust-valves being most rapid at the instant of release, the steam can be held to almost the end of the stroke, in consequence of the port opening to its full width at the instant the crank reaches the centre.

The eccentric is forged on the shaft, which gives compactness, exactness of construction, and prevents it being shifted by accident or design. The cylinder-heads are designed to receive the steam from the boiler, by which arrangement the clearance is reduced to a minimum, and the economy of the engine increased. Great pains are taken with the crank- and cross-head pins, which are made of the best steel, and hardened. The connecting-rod

boxes are made of gun-metal or bronze. The upper and lower sides of the cross-head wrist are made flat. The crank-pins are of unusual diameter in proportion to their length, which brings the flattened connecting-rods close up to the faces of the accurately balanced disc-cranks. The construction of the marine pillow-block bearings is the result of much study, and presents many improvements over those to be seen in ordinary engines. The fly-wheels of these engines, like the cranks, are turned and accurately balanced, which insures smooth motion in the revolving and reciprocating mechanism. The regulator employed on these engines is what is known as the Porter governor, and is peculiarly adapted to this class of engines; and, in consequence of being more powerful and sensitive than any other governor in the country, it has been successfully applied to the controlling of the valves of other engines which no other governor in the market would regulate.

The high-speed engine has not been heretofore appreciated in this country. As has been heretofore stated in this book, most intelligent American engineers entertain the idea that there is nothing to be gained by running an engine at such high velocity, or employing extraordinary high pressure, because an engine, to run at such an extraordinary speed, needs to be built with great care, of first-class and expensive material, which increases its first cost. Besides, the cost of maintenance of such a machine is a continual source of annoyance. It is a well-established fact in mechanism, that haste, beyond a certain limit, induces waste, and that any attempt to force any machine, steam-engines and boilers included, induces rapid wear, deterioration, and eventually the destruction of the machine. The high-speed engine is one of the innovations that at different times, in the opinion of their advocates, were going to revolutionize the whole system of steam engineering. But their impracticability soon became evident, and they died out, only to give place to another set of schemes that have proved equally delusive.

Questions.

What are the functions of the steam-engine indicator?

How would you proceed to attach the indicator?

Under what three heads may the particulars derived from an indicator diagram be classed?

State the conditions which are instrumental in determining the conformation of a diagram.

Explain the points of difference between diagrams taken from automatic cut-off engines and those taken from slide-valve throttling engines.

How would you draw the theoretical expansion curve geometrically?

How would you trace a theoretical compression curve?

From what circumstances does the inaccuracy of the theoretical curve arise?

Describe the adiabatic curve.

How would you locate the theoretical terminal pressure corresponding to the actual cut-off?

How do you account for the difference in theoretical correctness as shown by expansion curves of diagrams taken from different engines?

Why is the incorrectness of the expansion curve less with an engine heavily loaded than with a light load?

If the deviation of the expansion curve in diagrams (other things being equal) be found to be greatest when the water is high in the boilers, and the steam rapidly generated, to what cause might it be assigned?

What is the most obvious lesson to be deduced from the facts in our possession in regard to the incorrectness of the curves of diagrams taken from large engines?

What should be considered in indicator diagrams as indications of good construction and performance?

How do you calculate the mean effective pressure?

How would you space the ordinates?

How do you calculate the indicated horse-power?

How do you calculate the theoretical consumption of water from indicator diagrams?

How do you make allowance for clearance and cushion?

How do you estimate the effect of compression?

What is the object of indicator diagrams?

What information do indicator diagrams supply?

How do indicator diagrams show the steam-pressure in the cylinder?

When the line representing the back pressure in the diagrams of high-pressure engines shows more than one pound above atmosphere, or in low-pressure engines more than two or three pounds than the vacuum-gauge shows in the condenser, what does the diagram indicate?

How does the diagram show whether the valves are properly set or not?

How does the diagram show whether the piston and valves are leaking or not?

How does the diagram show whether the steam is throttled or not?

How does the diagram show the effect of small steam-ports and steam connections?

How does the diagram show the effect of exhaust lead?

How does the diagram show that the indicator is out of order?

How does a diagram show the point of cut-off?

How does a diagram show the state of the vacuum in the condenser, and whether too much injection-water is used or not?

Sketch a diagram, and explain it.

Show the points of excellence in a perfect diagram.

Show the steam, atmospheric, and vacuum lines and the expansion and exhaust curves.

Define the adiabatic curve, and explain how it is obtained.

Define the asymptote lines.

Define the term compression.

Define the term cushion.

Define the term clearance.

Define the terms flexure and contrary flexure, and demonstrate them on the diagram.

Define the term hyperbola, and illustrate it.

Give the meaning of the term isothermal.

Point out the ordinates on the diagram.

Define the term parallelism.

What is meant by the initial-pressure?

What is meant by the term mean effective pressure?

What is meant by the term terminal-pressure?

What is meant by the term scale in its application to the diagram?

Explain the functions of the spring in its relations to the indicator.

Explain the meaning of I. H. P., N. H. P., M. E. P., and H. P.

Explain the meaning of the term string.

Define the term undulating.

What are the use and functions of the dynamometer?

What is the meaning of the letters *B* and *T* which are frequently seen on diagrams?

Define the term zero when applied to indicator diagrams.

Give the formula for finding the horse-power of an engine from indicator diagrams.

What are the functions of the planimeter?

Explain the most correct method of using the planimeter.

What is the exponent of the work performed by a steam-engine? For the meaning of the term mean effective pressure, see page 259.

What is the best criterion of the most economical results which a steam-engine is capable of producing?

Before purchasing a steam-engine or other machinery, what considerations ought to be taken into account?

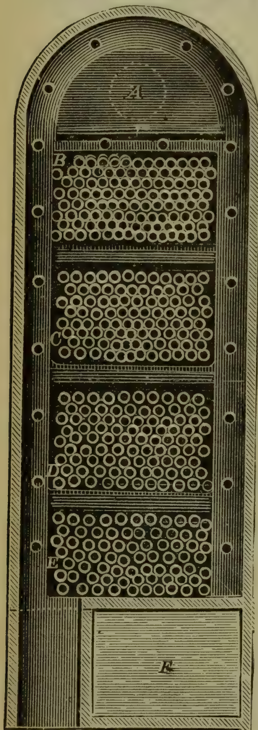
Is there any difference in the consumption of fuel required to carry steam at 100 lbs. pressure instead of 50 lbs. per square inch?

How should engines in factories be located?

PART FIFTH.

Condensers.

The condenser is one of the most necessary and important adjuncts of the low-pressure engine, as in the perfection of the vacuum produced in it by the condensation of the steam lies the economy of that class of machines. All the other parts of the engine may be modified, and many of them, in some cases, dispensed with, as in the trunk and oscillating engines. Even the air-pump, as has been shown in a former article, is not an absolute necessity; but, whatever changes a condensing engine may undergo, the presence of the condenser is an imperative necessity.



End View of a Surface Condenser with the Bonnet removed.

The two kinds of condensers in general use are known as the jet and surface. The surface condenser consists of an iron box, in which brass or copper tubes are inserted in tube sheets, similar to those of a tubular steam-boiler, through which the water is forced by the circulating pump, for the purpose of condensing the steam. In some cases condensation is effected by bringing the exhaust steam in contact with the outside of the tubes, the circulating water being on the inside; while in others the steam is exhausted into the tubes, and the circulating water distributed on the outside of them. There is no especial ad-

vantage in the former over the latter, nor *vice versâ*, the arrangement being only a matter of taste. In the proportioning of either surface or jet condensers there appears to be great latitude in practice, but in the surface condenser a certain amount of cooling surface must be provided; more than the required quantity is a waste of material, and incurs unnecessary weight and first cost, besides the extra time required to remove the air before starting. The objection to a small condenser is, that in case the air-pump should fail to operate properly, the condenser would soon become choked with water; the best guide in such cases is practical experience.

In the surface condenser, the cold water is lifted by a circulating pump through a pipe in the ship's side or bottom and forced through the tubes, and thence overboard. The number of times the water circulates, depends on the design and arrangement of the condenser. In some condensers it circulates once, in others twice, and in others three or even four times. In the cut on page 338 the steam enters at *A*, and the injection-water at *B*, which returns through section *C*, re-returns through section *D*, and is forced overboard through *E*. *F* represents the hot well containing the water of condensation, which is returned to the boilers by means of the boiler feed-pumps.

The capacity of the circulating pumps of the best class of surface-condensing, compound engines is about 1 to 22, or 1 cubic foot of circulating pump capacity to 22 feet in the low-pressure cylinder, and in the proportion of about 1 cubic foot of circulating pump capacity to 588 feet of cooling surface in the tubes of the condenser. The proportion of cooling surface in the best class of surface condensers is about 28 square feet of cooling surface to 1 cubic foot in the low-pressure cylinder, or 6½ square feet to the indicated horse-power.

The advantages of surface condensers are, that they furnish fresh water to the boilers, (since the sea injection-water does not mingle with the water of condensation, thus obviating the loss induced by scale in the boiler,) that steam of any pressure can be condensed, and that the vacuum is more perfect than in the jet

condensers. Their disadvantages are, extra weight and first cost, and that the tubes are liable to become leaky, and impair the vacuum. In case the tubes should become so leaky as to be beyond remedy, the surface condenser may be converted into a jet condenser by admitting the exhaust steam and injection-water above the tubes; but the jet condenser cannot be changed into a surface condenser under any circumstances.

The tubes of surface condensers are made of brass or copper, generally about $\frac{5}{8}$ of an inch in diameter. They were formerly riveted into the heads or tube sheets, but in consequence of the lightness of the material of which they are composed, and of its great limit of expansion, they soon became loose and leaky, as a result of which riveting was abandoned. They are now generally made tight by means of some fibrous material, such as cotton or India-rubber, but more recently by brushings of kiln-dried or condensed wood.

The jet condenser consists of an iron pot or shell, into which the steam is exhausted. The water rises through a pipe in the ship's side, by the pressure of the atmosphere, and is distributed by a rose, an arrangement similar to the nozzle of an ordinary garden watering-pot, which, as it frequently became choked by substances carried in by the injection-water, was abandoned. The distribution of the water is now effected by allowing it to strike a cone in the cover of the condenser.

The capacity of jet condensers may be from $\frac{1}{15}$ to $\frac{1}{20}$ the capacity of the steam-cylinder, or it may be of the same capacity as the air-pump. The advantages of a jet condenser are, that it is light, simple, and inexpensive; and its disadvantages, that the saline matter contained in the injection-water is carried into the boilers, which lessens the economy of fuel, and that steam of a very high pressure cannot be successfully condensed in it. Should the condenser become so impaired, as to be incapable of creating a vacuum, the connection between the condenser and the engine may be separated, and the engine allowed to exhaust into the atmosphere, when it becomes a non-condensing engine.

A snifting-valve is fixed on the condenser to allow the air and water to escape when the condenser is blown through. The vacuum in the condenser keeps it closed, and, in the event of a great head of water, or pressure in the condenser, the valve will ease up and allow it to escape.

TABLE

SHOWING THE FORCE WITH WHICH THE UNCONDENSED STEAM ARISING FROM THE WATER IN THE CONDENSER RESISTS THE ASCENT OR DESCENT OF THE PISTON, ACCORDING TO ITS TEMPERATURE.

Temperature, Fahrenheit.	Force in Inches of Mercury.	Pounds per Square Inch.	Temperature, Fahrenheit.	Force in Inches of Mercury.	Pounds per Square Inch.
32	0·200	0·100	130	4·36	2·17
40	0·250	0·128	135	5·07	2·52
50	0·360	0·181	140	5·77	2·88
55	0·416	0·215	145	6·60	3·28
60	0·516	0·260	150	7·53	3·74
65	0·630	0·311	155	8·50	4·22
70	0·726	0·361	160	9·60	4·76
75	0·860	0·428	165	10·80	5·37
80	1·01	0·505	170	12·05	6·04
85	1·17	0·585	175	13·55	6·75
90	1·36	0·680	180	15·16	7·58
95	1·58	0·805	185	16·90	8·47
100	1·86	0·900	190	19·00	9·50
105	2·10	1·07	195	21·10	10·58
110	2·53	1·26	200	23·60	11·81
115	2·82	1·43	205	25·90	13·01
120	3·30	1·50	210	28·88	14·43
125	3·83	1·902	212	30·	15·

The temperature of the water in the hot wells of surface-condensing engines is generally about 100° to 110° Fah. A higher temperature would affect the vacuum and injure the air-pump

valves, while a lower temperature would cool the cylinder, and cause a waste of fuel by the condensation of the steam. A very low temperature causes increased consumption of fuel, while a very high one causes a loss of power, owing to the back pressure induced by the uncondensed vapor in the condenser, which will be shown by the vacuum-gauge.

In the jet condenser, when the bilge-injection is opened, the air-pump draws off the air from the pipe, when the air in the ship, pressing on the surface of the bilge-water, forces it up the pipe into the condenser. In the surface condenser the circulating pump creates the vacuum, and the air presses the water up.

In a jet condenser, if the injection-water is not shut off when the engines are stopped, the condenser will be filled with water, and, if not cleared before the engine is started, may cause serious damage to the cylinder or condenser.

Relative Quantity of Injection-Water Required to Condense a Certain Volume of Steam.

The weight or quantity of injection- or condensing-water required for a given weight or volume of steam depends upon several conditions: 1. The pressure at which the steam is exhausted. 2. The absolute pressure existing in the condenser after the vacuum has been formed. 3. The temperature at which the injection-water enters the condenser. While the first and second conditions vary but slightly with uniform load and steam pressure, the third will vary with the season, and even with the weather; consequently, more condensing-water is required in summer than in winter. But the average amount may be illustrated as follows:

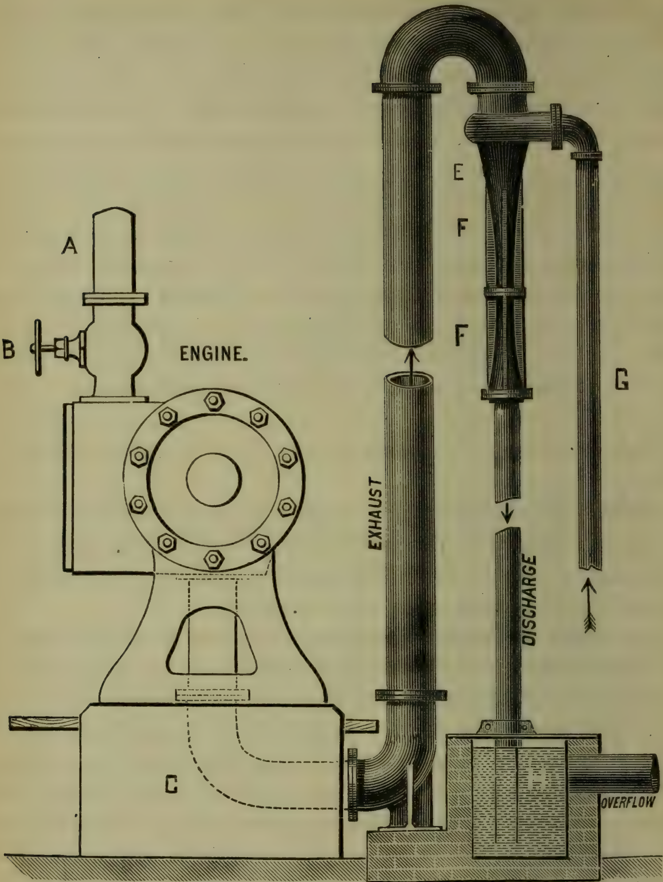
Example.— Suppose the pressure in the cylinder at release or exhaust be 5 lbs. above atmosphere, and the absolute pressure in the condenser, after vacuum is formed, be 2 lbs., corresponding to a vacuum of 26 inches of mercury. Each pound of steam exhausted at 5 lbs. above atmosphere contains 1183 thermal units,

and the thermal units per pound of condensation at 2 lbs. absolute pressure in the condenser are 126. Hence the thermal units to be absorbed by the condensing-water are $1183 - 126 = 1057$. Suppose the temperature of the injection-water be 80° Fah., then each pound of condensing-water takes up $126 - 80 = 46$ thermal units; and pounds of condensing-water per pound of steam condensed become $\frac{1057}{46} = 23$ nearly. Suppose, again, that the temperature of the injection-water be 40° Fah., then each pound of condensing-water takes up $126 - 40 = 86$ thermal units, and pounds of condensing-water per lb. of steam condensed are $\frac{1057}{86} = 12 \cdot 3$ lbs., from which it will be seen that the average proportion of condensing-water to the water of condensation is about as 18 to 1.

Rule for finding the cooling surface in the tubes of surface condensers.

Multiply the circumference of one tube in inches by its length in inches; then multiply that product by the whole number of tubes, and divide by 144. The quotient will be the number of square feet of cooling surface in the tubes.

The tubes of surface condensers frequently become foul on the inside, owing to the grease used to lubricate the cylinder being carried over with the exhaust steam, and they become foul on the outside by the saline matter in the injection-water adhering to them. There are various ways of cleansing them, one of which is to admit steam of a high pressure from the boiler to the condenser. Another is to fill the condenser with a strong solution of soda of potash, but in both cases care must be taken not to destroy the packing round the tubes, of whatever material it may be composed. If the tubes are packed with wood, and allowed to become too hot by the action of the steam, they will become charred and drop out; if packed with India-rubber, they will be destroyed by too high a temperature; if the solution of potash used to clean them is too strong, they are liable to be ruined.



The Injector Condenser.

A shows the steam-pipe; *B*, the stop-valve; *C*, the exhaust-pipe; *E*, the annular head into which the condensing-water is thrown through the pipe, *G*, and by the arrangement of which the water is formed into a sheet; *F F* shows the two inverted nozzles through which the condensing-water escapes into the hot well, *H*.

The Injector Condenser.

The cut on the opposite page represents the injector condenser, so called because its action is similar to that of the Giffard boiler feed-injector. It consists of two conoidal nozzles, joined by a straight neck, and swelled at the upper end for the junction of the water-nozzle. Within is the exhaust-steam nozzle, which forms within the condenser a narrow, annular space for the entrance of the condensing-water. The sides of the condenser (which are paraboloidal curves) are smoothly finished, as is the contracted neck below, to diminish the resistance of the water. When used in connection with a condensing engine, the air-pump may be dispensed with, as steam of atmospheric pressure will flow into a vacuum at the rate of 1600 feet per second.

When the exhaust steam from the engine meets the thin film of water which enters by the annular space, it is instantly condensed. As the water passes down, the contracting outline of the condenser gradually brings it to a solid jet in the neck below, through which it rushes with a velocity due to its pressure. The air which has entered the condenser with the water, or through leaky joints or stuffing-boxes, together with the uncondensed vapor, is thus drawn down into a contracting, hollow cone of water, until finally expelled through the neck. This latter being straight for a distance, is virtually the air-pump, having a solid piston of water moving at a high speed; thus is the steam condensed, and the vacuum formed by a single process, and with greater certainty than in any other way. The air and vapor having passed the contracted neck, enter the tapering nozzle below, and expanding therein are prevented from returning to the condenser above.

The method of operation of the injector condenser when the engine is started is as follows: the exhaust steam expels the air from the exhaust-pipe and condenser; then a jet of cold water from a pump or tank creates a vacuum, which may be maintained by a head of water of 10 feet fall. The discharge-water passes off at

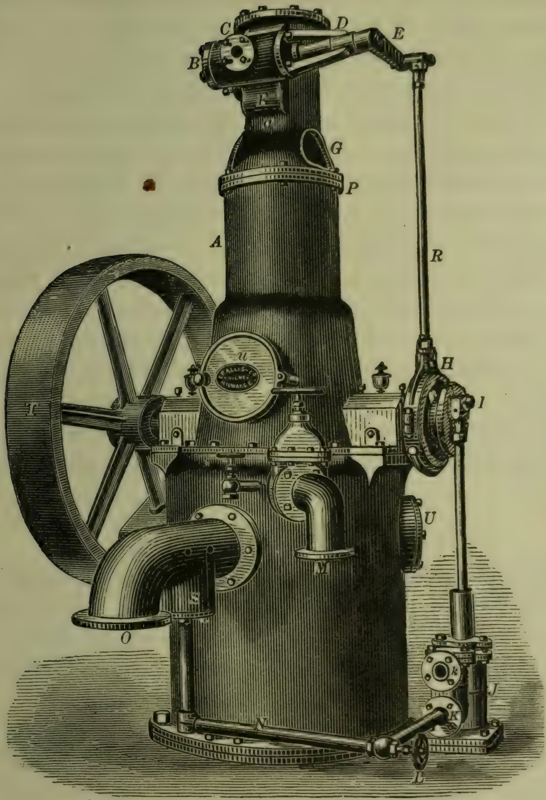
a temperature of 110° to 112°, when the vacuum is equal to 26 inches of mercury.

The advantages of the injector condenser are, that it is cheap, light, simple, and durable; that the injection-water is self-regulating; that there is no possibility of the water being carried over into the cylinder; that, simply by the removal of a bonnet, the exhaust steam may be allowed to escape into the atmosphere, thus converting the engine from low to high pressure; that the condenser requires no foundation or any other attachment save an ordinary pump to raise the water when there is not sufficient fall; and that the condenser can be attached to any engine in any locality where the necessary supply of water can be obtained, which ranges from 22 to 25 times as much as that from which the steam would be generated. It is claimed that a saving of over 20 per cent. may be made where these condensers are used, while the first cost is trifling.

Independent Condenser and Air-Pump.

The annexed cut represents a very ingenious and convenient combined condenser and air-pump, designed by Edwin Reynolds, and manufactured by E. P. Allis & Co., Milwaukee, Wisconsin, to be used in connection with the Reynolds Corliss engine, and which can, in case of accident to the engine which drives the machinery, be used as an independent or auxiliary engine, which may be used while making repairs. A foot-valve is entirely dispensed with, and only bucket- and delivery-valves used. They are fitted up either with or without a steam-cylinder, and when the latter is dispensed with the air-pump is driven by a belt from the engine to the wheel, *T;* but when driven by the steam-cylinder a regulator is employed, which is not shown in the cut. The barrel, *A*, which is bored out on a line with the steam-cylinder, *a*, forms a cross-head guide, to which the steam-piston is attached. Four rods form a connection with the cross-head of the air-pump below the crank-shaft.

The head, B, shows the bonnet or cover of the valve-chamber,



Independent Condenser and Air-Pump.

Designed for the Reynolds Corliss Engine, and Manufactured by E. P Allis & Co., Milwaukee, Wis. See description on pages 346 and 348.

and the flange, *C*, the steam-pipe connection. *D* represents the bracket containing the valve-stem stuffing-box, the outer end of which forms a bearing for an oscillating valve, to the stem of which the crank, *E*, is keyed, which operates the steam-valve. The swell, *F*, constitutes the steam-ports from the valve-chamber to the end of the steam-cylinder. *G* shows an opening provided for getting at the stuffing-box of the piston-rod and oiling the cylindrical cross-head. The eccentric, *H*, gives motion to the steam-valve through the rod, *R*, and crank, *E*. The crank *I*, which is slotted for the purpose of adjusting the length of the stroke, drives the force-pump, *J*, which may be used either as a lift and force, circulating, or boiler feed-pump. *k* represents the delivery- and *K* the suction-pipe. The pipe, *M*, is the injection-pipe; *N*, the suction for the force-pump, which takes its water from the pocket, *S*. The valve, *L*, is used to regulate the quantity of water admitted to the force-pump. The pipe, *O*, is the overflow from the air-pump, which is in the centre of the condenser. The plate, *V*, covers a man-hole, through which access is had to the valves and piston of the air-pump.

The Vacuum.

If the cylinder of a steam-engine be filled with vapor, it cannot be said to be void of matter; but if the vapor is condensed, and the water from which it was vaporized drawn off, there would be created in the cylinder what is termed a "vacuum," or void space. The absolute pressure of steam is measured from zero, or perfect vacuum, and consists of the pressure indicated by the steam-gauge (which is known as pressure above atmosphere), added to the pressure of the atmosphere, as shown by the barometer. The latter is, for all practical purposes, a constant quantity for any given locality,* and may be roughly taken at 14·5 lbs., corresponding to 29·50 inches of mercury. Vacuum-gauges are usually graduated to agree with the scale of the barometer, and the vacuum is usually stated in inches of mercury. To the steam-

* See Table on page 498.

pressure, as indicated by the gauge, add 14·5 lbs. for total pressure. Thus, if the pressure by gauge is 60 lbs., the total pressure is 74·5 lbs. Consequently, the total pressure on the steam side, at any point in the stroke of the piston, is the pressure above the atmosphere plus 14·5 lbs., and the total pressure for the whole stroke is the mean-pressure above the atmosphere plus 14·5 lbs. Thus, if the mean-pressure for whole stroke is 30 lbs., the total mean-pressure is 44·5 lbs.; and this 44·5 lbs., whether the engine is condensing or non-condensing, is the variable factor in estimating the load on the engine. Now, if the engine be non-condensing, the 14·5 lbs. (pressure of the atmosphere) on the steam side of the piston is balanced by an equal atmospheric pressure on the exhaust side, and its effect is neutralized; but if the engine be condensing, a large proportion of the pressure of the atmosphere on the exhaust side of the piston is removed, and an equivalent portion of the pressure of the atmosphere on the steam side of the piston made to do useful work. With well-proportioned condensing apparatus, the pressure of the atmosphere on the exhaust side of the piston can be reduced nearly 90 per cent.

TABLE

SHOWING VACUUM IN INCHES OF MERCURY AND POUNDS PRESSURE PER SQUARE INCH.

Mercury.	Pounds.	Mercury.	Pounds.
2·037	1	16·300	8
4·074	2	18·337	9
6·111	3	20·374	10
8·148	4	22·411	11
10·189	5	24·448	12
12·226	6	26·485	13
14·263	7	28·552	14

The lower the temperature of the water leaving the condenser, the better the vacuum, and the more conducive to power, always

supposing there be no air-leaks. Watt found a temperature of 100° in the water leaving the condenser more beneficial than 70° or 80°, supposing there be an abundant supply of cold water. It may be explained in this way. A better vacuum due to a temperature of 70° or 80° requires so much cold water in the condenser, (which must afterwards be pumped out against the pressure of the atmosphere,) that the gain in the vacuum does not equal the loss of power caused by the additional load on the pump. There is, therefore, a clear loss by the reduction of the temperature below 100°, if such reduction be caused by the admission of an additional quantity of water.

The vacuum is maintained in the condenser by the action of the air-pump. A perfect vacuum cannot exist, and in the condenser of an engine there is always more or less pressure from imperfect condensation, and air passing in with the condensing-water.

The vacuum is measured by inches in the height of a column of mercury, 2 inches of mercury equalling one pound pressure per square inch; thus, 20 inches of mercury means 10 lbs. pressure per square inch. If the steam-gauge shows 10 lbs. pressure, and the vacuum-gauge registers 20 inches, there is a vacuum equal to 10 lbs. per square inch in the condenser.

The vacuum is maintained in the condenser by the exhaust steam being constantly condensed, by either mixing with the cold injection-water or by being brought in contact with the cooling surface of the tubes in the surface condenser.

A vacuum is produced in a condenser by the steam (when it first enters) driving out the air; and, when condensed into water, it occupies 1669 times less space than it did before being condensed, as 1700 cubic feet of steam produce one cubic foot of water.

To produce a vacuum in a jet condenser, open the blow-through valve, when the steam, in its passage through, will blow out all air and water in the condenser; and as soon as the steam issues from the snifting-valve the blow-through valve may be shut, and the injection-cocks opened, when the cold water mixing with the steam forms a vacuum. When the gauge shows a sufficient vac-

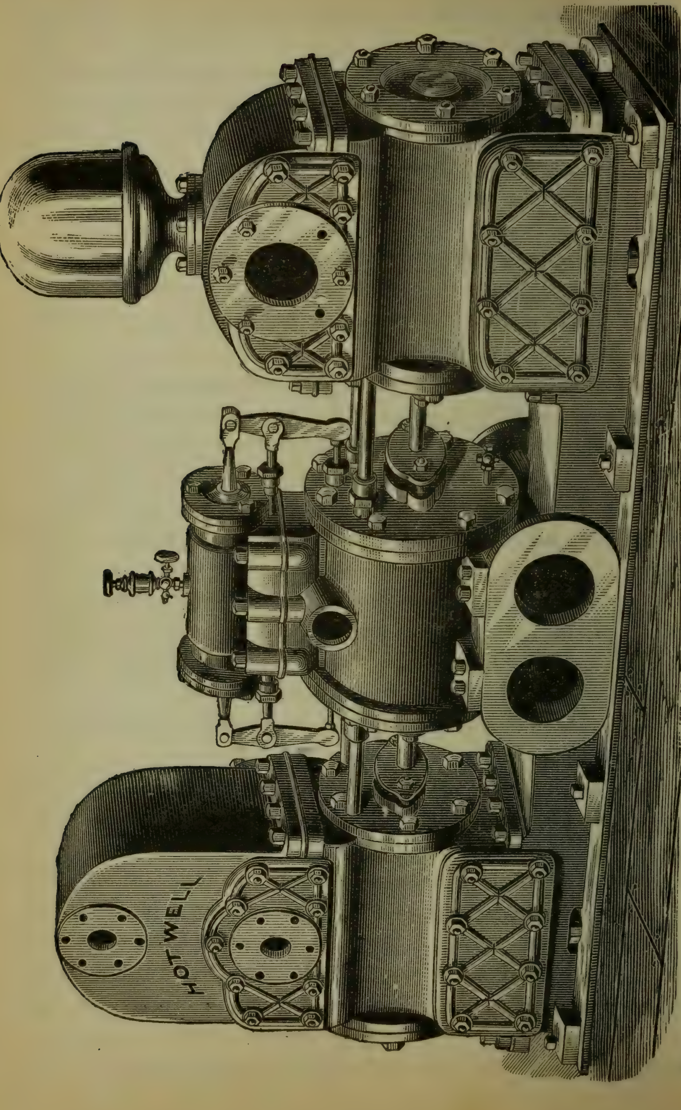
uum, shut the injection-cocks, in order to prevent the condenser from being flooded.

To produce a vacuum in a surface condenser, open the injection-valve shortly before starting the engine, so that the circulating-water may enter the condenser tubes, and cool them. Then, when the engine is started, the exhaust steam comes in contact with the cooling surface of the tubes, and is condensed when a vacuum is formed.

If the steam-gauge shows 60 lbs. pressure, and the vacuum-gauge 26 inches, it means that there are 60 lbs. of steam pressing on one side of the piston, and 13 lbs. of resistance removed from the other side.

The state of the vacuum is shown by the vacuum-gauge attached to the condenser; and, if it be imperfect, the cause must be ascertained and the fault corrected. If the water in the hot well be above the ordinary temperature, more injection-water must be admitted; and, if the vacuum continues imperfect, the cause may be due to an air-leak, the location of which the engineer must endeavor to discover. Very often the fault will be found in the valve- or cylinder-cover, which must then be screwed down more firmly; or in the joint of the eduction-pipe, the gland of which will require to be tightened. The door of the condenser should also be examined. The joints of the condenser may be tested by holding a candle to them, the flame of which will be drawn in if the joints are leaky.

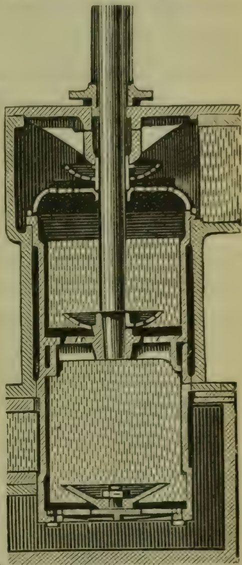
A vacuum is not power, as all power in the steam-engine is derived from the pressure of the steam on the piston; if there is no resistance on one side of the piston, the entire pressure on the other side is available. Whenever there is resistance on one side of the piston, it must be deducted from the pressure on the other side. There is hardly such a thing as a perfect vacuum. The philosopher Torricelli asserted correctly that nature abhors a vacuum; consequently, if a perfect vacuum can be attained, it cannot be maintained long, as the wear of the machinery, the packing around the air-pump rod, and other causes, contribute to impair it.



Independent Air- and Circulating-Pump, with Air-Pump at one end, Circulating-Pump at the other, and Steam-Cylinder in the middle.

Air-Pumps.

All condensing engines have of necessity to be provided with an air-pump, for the purpose of extracting the air, injection-water, and the water of condensation from the condenser, in order to maintain a vacuum. There does not appear to be any uniform, recognized rule, among marine engineers or manufacturers of surface-condensing compound engines, for proportioning the air-pumps to the steam-cylinder, as, while some builders make the capacity of their air-pumps one-eighth of that of the low-pressure cylinder, others make it one-tenth, and others one-eleventh; the average of the number examined being about one-ninth.



Section of a Marine Air-Pump.

The air-pumps of the steamships Pennsylvania, Ohio, Indiana, and Illinois, of the American Line, are one-eleventh the capacity of the low-pressure cylinder. And, as these engines have the reputation of being very economical, it should be presumed that their proportions are good; nevertheless, they are evidently too large, as one-fifteenth would be nearer to a correct proportion. The tendency among marine engineers is to overdo the thing in the case of air-pumps, perhaps under the impression that a large air-pump creates a better vacuum, and, as a result, air-pumps of enormous diameter and long stroke are attached to marine engines; whereas, the air-pump has very little to do with the vacuum, its functions being simply to clear the condenser of water and air. Any proportion that will accomplish this will fulfil all the necessary requirements. An air-pump too large for the purpose for which it is intended, can have no other effect than to absorb much of the power which might be utilized in in-

creasing the speed of the engine and economizing the fuel. The air-pump piston, being resisted by the pressure of the atmosphere, absorbs from four to five per cent. of the power of ordinary simple condensing engines, and from two to three per cent. in the better class of compound marine engines; the power required to work it being greatest when the vacuum is most perfect, and least when the vacuum is impaired. A good deal also depends on the mechanical arrangement employed to work it, as well as on the condition of its packings, bearings, proportions, etc.

The capacity of the air-pumps of condensing engines using a jet or spray, ranges from one-fifteenth to one-twentieth the capacity of the cylinder. As it requires from 22 to 30 times as much water to condense steam as there is water in it (according to the pressure and temperature), the air-pumps ought to be proportioned to meet the maximum demands. The right proportions of air-pumps for both jet- and surface-condensing engines may be found by *calculating* the displacement of the steam-piston, and that of the air-pump for one minute, and *dividing* the former by the latter. The use of the air-pump in connection with condensing engines, as before stated, is not an absolute necessity in all cases, as, with a head of water having a fall of about 13 feet, a vacuum can be formed and maintained in the condenser without an air-pump, providing the end of the delivery-pipe is submerged in a tank of water.

Vertical air-pumps, with valves in their pistons or buckets, give the best satisfaction, as, in that case, the air and vapor are lifted and forced out of the condenser, relieving the exhaust and increasing the vacuum. The capacity of the openings through the valve-seats of air-pumps should be such that the maximum flow of the water through them will not exceed 10 feet per second. For instance, suppose a pump of 12 ins. stroke to make 50 strokes per minute, the maximum travel of the bucket at midstroke will be about 2·6 feet per second. Then, as $10 \div 2{\cdot}6 = 3{\cdot}84$, the capacity of the opening should not be less than one-fourth the area of the pumps.

Air-pumps are frequently very injudiciously located, being placed above the condenser; whereas, if placed below it, their requirements would be fewer, as the water would fall by gravity from the condenser into the air-pump. In some cases the air-pump extends down through the condenser, so that the openings are nearly on a level with the bottom of the condenser, which is a good arrangement in every respect, except that it necessitates a long stroke, which has a tendency to absorb power.

Independent air-pumps, a cut of which may be seen on page 352, having an air-cylinder at one end, the circulating-water cylinder at the other, and the steam-cylinder in the middle, are being very generally adopted on ocean steamers. The claim set up for them is, that, as they are independent of the engine, they can be worked faster or slower, according to the circumstances of the case; that they absorb none of the power of the engine, and are freer from liability to accident in stormy weather, or whenever the engine races, than air-pumps attached to the main engine; that they can be started, and a vacuum formed, before the engine commences to work; that the injection-water can be more easily regulated; that they require no expensive foundation; that, in consequence of the water- and air-pistons being on each end of the steam-piston, they have a more steady and uniform motion than the ordinary air-pump has, and that, in consequence of all their parts being accessible, they can be easily examined, and any derangement remedied or readjusted, without interfering with the working of the engine.

In a surface-condensing engine, the air-pump has only to extract the water resulting from the condensed steam and the uncondensed vapor from the condenser. In a jet-condensing engine, the air-pump has to withdraw both the injection-water and the water of condensation; the work to be performed by the latter being from 25 to 30 times greater than that of the former.

An air-valve is sometimes fitted to a circulating, reciprocating, or double-acting pump, for the purpose of admitting air to the

water on the up stroke. As the valve is closed against the down stroke, the air admitted serves to soften the shock of the water.

A bucket air-pump is a single-acting pump, being, in fact, a piston with a valve fitted to it, which closes on the up stroke and opens on the down, lifting a quantity of water equal to its capacity at each stroke of the engine.

A piston air-pump is a double-acting pump, the piston being solid. It is fitted with suction- and delivery-valves, and discharges with each stroke.

A plunger air-pump is a double-acting pump, resembling the bucket air-pump, except that it has no head-valves, and that the bucket-rod is fitted with a plunger. The effect of this is, that the plunger, owing to its displacement, discharges on both the up and the down stroke.

The double-acting air-pump has both suction- and delivery-valves; but it is possible with the single-acting pump, in some cases, to dispense with either the one or the other. They are generally made with pistons, though sometimes with plungers.

An air-pump with a foot-valve and no discharge-valve would be most affected by a leaky stuffing-box; and, while the foot-valve remains, the pump will draw water, but if removed, it will fail to work.

An air-pump trunk is a hollow cylinder attached to the bucket or piston, and working through a stuffing-box. Such an arrangement is rendered necessary when the pump is worked directly off the crank-shaft, or where it is located so close to the levers, through which the motion is transmitted from the engine, as to render the appliance of an intervening cross-head and links impossible. The difference in the discharge is equal to the relative difference between the displacement caused by an ordinary air-pump rod and that caused by the trunk.

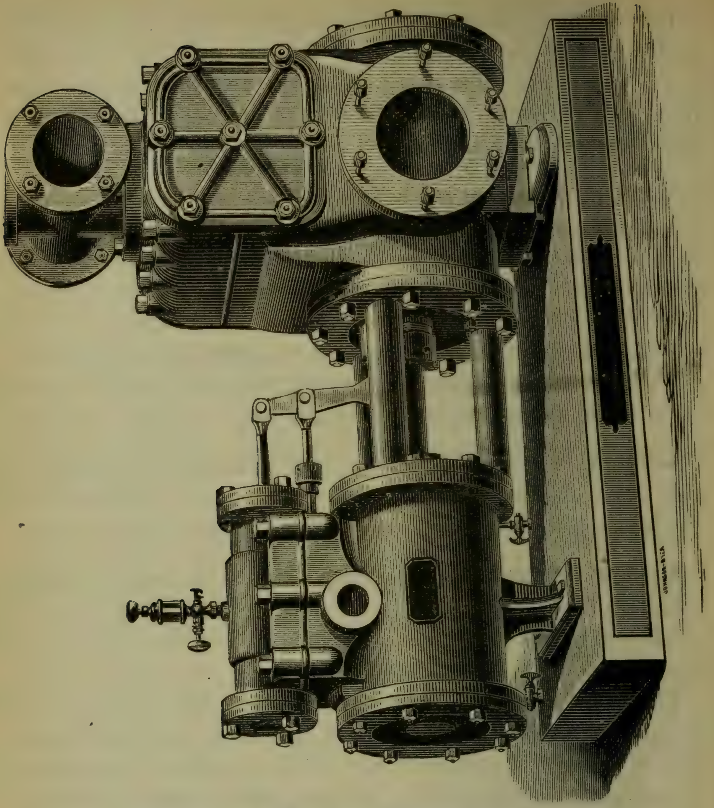
The air-pump pet cock or valve is generally placed below the head-valve and above the bucket. It opens with the down stroke of the pump, and admits air to act as a cushion on the water. When the delivery-valve is opened, the engineer can tell by its action whether the pump is working properly or not.

An air-pump bucket is a hollow piston, generally made of brass, with a grating in the top, and a boss (water-tight) which receives the rod in its centre, from which strengthening ribs run to the rim of the bucket. The outside of the bucket is grooved to receive water-tight packing. The valves, which are generally of India-rubber, and whose lift is regulated by a guard secured by a nut, and against which the valve is pressed when the bucket is on the down stroke, are on the top of the grating.

Air-pump rods are generally made of wrought-iron, and covered with a skin of Muntz metal, or brass, to prevent the oxidization to which wrought- or cast-iron rods are exposed.

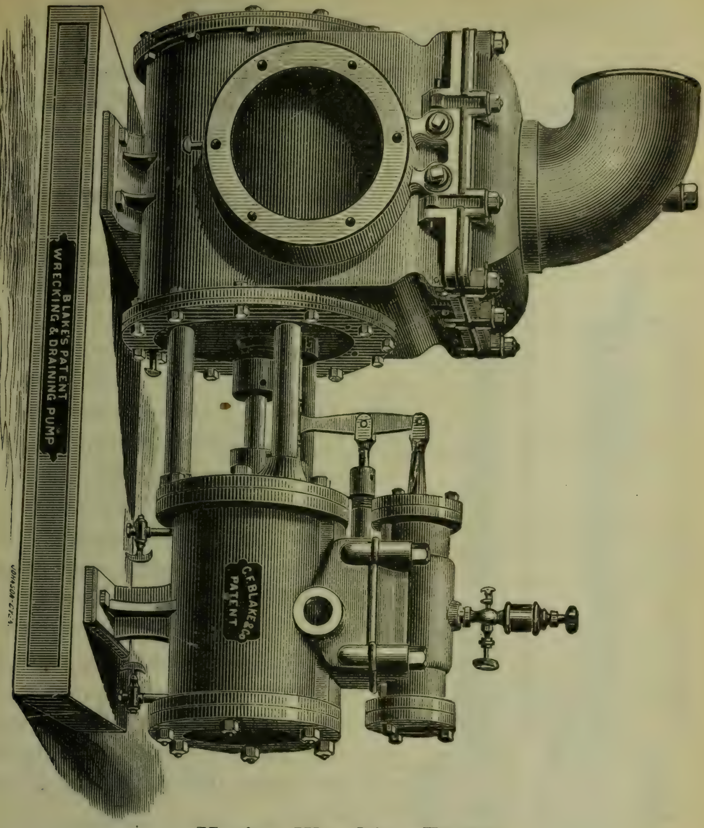
A ship's side air-pump discharge-valve is generally a mitred valve, with its spindle passing through a gland in the cover, on which a weight is placed to keep it shut. It differs from a stop-valve in having a lift and weight.

There are numerous contrivances in use for dispensing with the air-pump, such as the injector condenser, which produces a sheet of water in the exhaust-pipe; but the necessary arrangements for operating them generally cost more than a good reliable air-pump, though the first cost of the former is less than that of the latter. Besides, the vacuum is never so perfect when produced by any such arrangement as when created by a close condenser and air-pump. This becomes obvious, since we know that, even with the most perfect mechanism, it is almost impossible to attain a perfect vacuum, and maintain it for any length of time, as nature abhors a vacuum, as the atmosphere on the outside of a vessel is constantly endeavoring to equalize any unbalanced pressure that may exist on the inside.



Independent Marine Circulating-Pump.

Steam-Cylinder.	Water-Cylinder.	Stroke.	Gallons per Stroke.	Capacity per Minute, at Ordinary Speed.	Steam-Pipe.	Exhaust-Pipe.	Suction-Pipe.	Discharge Pipe.
12	6	12	2·61	100 strokes, 261 gal.	$1\frac{1}{2}$	$2\frac{1}{2}$	5	$3\frac{1}{2}$
14	9	12	3·30	100 " 330 "	2	3	5	$3\frac{1}{2}$
14	10	12	4·08	100 " 408 "	2	3	5	$3\frac{1}{2}$
10	9	18	4·96	70 " 347 "	$1\frac{1}{4}$	2	6	4
12	10	18	6·12	70 " 428 "	$1\frac{1}{2}$	$2\frac{1}{2}$	8	6
14	12	18	8·80	70 " 616 "	2	3	8	6
$14\frac{5}{8}$	12	24	11·75	50 " 588 "	2	3	8	6
$14\frac{5}{8}$	10	36	12·24	35 " 428 "	2	3	8	6

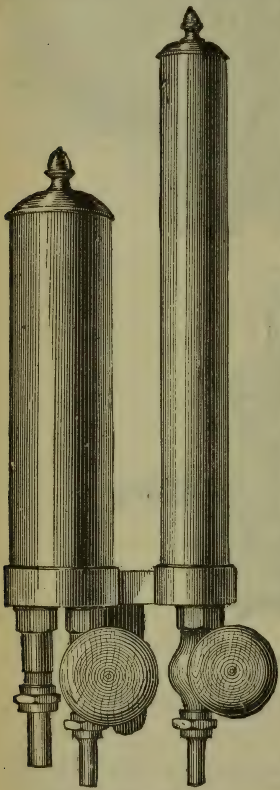


Marine Wrecking-Pump.

Steam-Cylinder.	Water-Cylinder.	Stroke.	Gallons per Stroke.	Capacity per Minute, at Ordinary Speed.		Steam-Pipe.	Exhaust-Pipe.	Suction-Pipe.	Discharge Pipe.
6	3¾	7	·33	125 strokes,	42 gal.	¾	1	2	1½
7¼	4½	10	·69	100 "	69 "	1	1½	2½	2
7¼	7	10	1·66	100 "	166 "	1	1½	4	3
8	5	12	1·02	100 "	102 "	1	1½	3½	3
8	8	12	2·61	100 "	261 "	1	1½	5	3½
10	6	12	1·47	100 "	147 "	1¼	2	3½	3
10	10	12	4·08	100 "	408 "	1¼	2	5	3½

The Salinometer.*

A Salinometer is a form of hydrometer used to determine the quantity of salt contained in the water of marine boilers, and by which the amount of water necessary to be blown out, to keep the water in the boilers at a certain density, may be ascertained. It is a graduated glass tube, and floats in the water at a height proportional to its density or saltness. It is marked 0 for fresh water; $\frac{1}{32}$ for sea-water that contains 1 lb. of salt to 32 lbs. of water; $\frac{2}{32}$ when it contains 2 lbs. of salt to 32 lbs. of water, and so on. Each division is subdivided into four parts, showing halves and quarters. It is graduated for a temperature of 200° Fah. A uniform standard of temperature is necessary, since water must be taken from the pressure in the boiler, in order that it may assume its regular temperature under the pressure of the atmosphere, because steam of different pressures has different temperatures, and a difference in temperature will alter the indications of the hydrometer.



The Salinometer.

The amount of salt in the water of a boiler may be ascertained by observing the degree of the boiling-point by means of a thermometer. To do this, a sufficient quantity of the water in the boiler should be drawn off in a long copper vessel, and brought to the boiling-point. Then immerse the thermometer. For every pound of salt contained in 32 lbs. of water, the temperature rises one degree. Thus, if the

* See page 651.

water contains $\frac{1}{32}$ of salt, it will boil at 213°; if $\frac{2}{32}$, at 214°; if $\frac{3}{32}$, at 215·5°, and $\frac{4}{32}$, at 216·6°.

Salt-water, at the usual density, contains $\frac{1}{32}$ of its weight of salt; consequently, if one pound of salt enters the boiler with every 32 lbs. of water, and 16 lbs. of that water be evaporated, the one pound of salt remains in the proportion of 1 : 16. Again, if $\frac{1}{2}$ of the 16 lbs. of water remains to be evaporated, the one pound remains in the 8 lbs. of water. Now, if these 8 lbs. of water were blown out of the boiler, the salt would go with it; and so long as that proportion is carried out, the saturation cannot exceed $\frac{4}{32}$; from which it is clear that, to keep water at $\frac{4}{32}$, one-fourth must be blown out; one-third at $\frac{3}{32}$, and at $\frac{2}{32}$ one-half of the water used for feed must be blown out.

The errors in the hydrometer may be corrected in the following manner: Every 10° difference in temperature will vary the indications $\frac{1}{8}$ of $\frac{1}{32}$, 200° Fah. being the standard. Then, if the water be 10° over 200° Fah., it will show $\frac{1}{8}$ of $\frac{1}{32}$ less than its true density; and if 10° below 200° Fah., it will indicate $\frac{1}{8}$ of $\frac{1}{32}$ more. Moreover, if the grade be 200° Fah., the thermometer shows 210°, and the hydrometer indicates a density of $\frac{2}{32}$, the true density will be $2\frac{1}{8}$; and if the temperature be 190°, it will be $1\frac{7}{8}$.

A Salinometer may be constructed by taking a long glass tube, and inserting in it sufficient shot to sink it in fresh water, marking the point at which the water stands in the tube. Then immerse the tube in water containing $\frac{1}{32}$ part of salt, when the point at which the water stands will be the sea-water mark. Similarly immerse in water containing $\frac{2}{32}$, $\frac{3}{32}$, etc., up to $\frac{12}{32}$ of its weight of salt, marking off the respective points at which the water stands. Transfer these marks to a scale, and paste it inside the bottle in exactly the same position as the marks on the bottle, and the result is a good salt-gauge. The temperature must always be the same as when the hydrometer was graduated.

How to use a Salinometer.— Draw off some water from the boilers, and when the ebullition has ceased, try its temperature with a thermometer. If the temperature exceeds that marked on the salinometer, let it cool till it reaches that degree; and if the tem-

perature is less than that marked on the salinometer, it must be raised till it reaches that degree. Then immerse the salinometer in the water and let it float; if the level of the water is at $\frac{2}{32}$ or less, there is no occasion for blowing off; but if it exceeds $\frac{2}{32}$, the water must be changed. The degrees of temperature usually marked on the salinometer are 190°, 200°, 210°. Before using the salinometer, it should be wet all over with water.

TABLE

SHOWING THE PROPORTION OF SALT IN THE WATER OF DIFFERENT SEAS.

	PARTS IN 1000.		PARTS IN 1000.
Baltic Sea	$6{\cdot}60 = \frac{1}{152}$	Mediterranean Sea	$39{\cdot}40 = \frac{1}{25}$
Black Sea	$21{\cdot}60 = \frac{1}{46}$	Atlantic at Equator	$39{\cdot}42 = \frac{1}{25}$
Arctic Sea	$28{\cdot}30 = \frac{1}{35}$	South Atlantic	$41{\cdot}20 = \frac{1}{24}$
Irish Sea	$33{\cdot}76 = \frac{1}{30}$	North Atlantic	$42{\cdot}60 = \frac{1}{23}$
British Channel	$35{\cdot}50 = \frac{1}{28}$	Dead Sea	$385{\cdot}00 = \frac{11}{28}$

TABLE

SHOWING THE BOILING-POINT OF SALT-WATER AT THE DIFFERENT DEGREES OF DENSITY, WHEN THE BAROMETER STANDS AT 30 INCHES.

	SATURATION.	BOILING-POINT.
Fresh water	....	212 ° Fah.
Sea-water	$\frac{1}{32}$	213·2 "
"	$\frac{2}{32}$	214·4 "
"	$\frac{3}{32}$	215·5 "
"	$\frac{4}{32}$	216·7 "
"	$\frac{5}{32}$	217·9 "
"	$\frac{6}{32}$	219·1 "
"	$\frac{7}{32}$	220·3 "
"	$\frac{8}{32}$	221·5 "
"	$\frac{9}{32}$	222.7 "
"	$\frac{10}{32}$	223·8 "
"	$\frac{11}{32}$	225·0 "
"	$\frac{12}{32}$	226·1 "

The meaning of the term saturation, in its relation to the water of marine boilers, means the quantity of salt it contains per gallon.

Saturation at $\frac{1}{32}$ means 4 oz. salt to one gallon fresh water; $\frac{2}{32}$, 8 oz. salt to one gallon water; $\frac{3}{32}$, 12 oz. salt to one gallon water, and so on. In carrying the water at $\frac{3}{32}$, twice as much is converted into steam as is blown off. At $\frac{2}{32}$, the water blown off and that converted into steam are equal. At $\frac{1\frac{3}{4}}{32}$, the water converted into steam equals $\frac{3}{4}$ of the water blown off.

The following table shows the method of regulating the saturation. 600 gallons of water, which is supposed to contain 7200 oz. of salt, being made the basis of the calculation.

	Water in Gallons.	Salt in Ounces.	
	600	7200	
Blown out . .	200	2400	
Fed in at $\frac{1}{32}$ to make	400	4800	
up for deficiency .	200	800	
	600	5600	
	200	steam	$\frac{1}{3}$ evaporated.
	400	5600	
Fed in . . .	200	800	
	600	6400	
	200	steam	$\frac{1}{3}$ evaporated.
	400	6400	
Fed in . . .	200	800	
	600	7200	

The following calculation shows the loss induced by blowing off as well as the gain derived from fresh-water condensers, pro-

viding they are tight, and the condensation of the steam be perfect. The degrees of heat imparted to the water converted into steam are the total heat of the steam minus the degrees of heat in the feed-water. The heat lost by blowing off is the difference between the heat of the feed-water and the sensible heat of the steam.

Rule for finding the percentage of loss induced by blowing off to prevent saturation.

Multiply loss by blowing off by 100, and *divide* the product by the total degrees of heat imparted to the water plus the heat lost by blowing off. (Observe that for $\frac{3}{32}$, as twice as much water is converted into steam as is blown off. For $\frac{2}{32}$, the amount is equal. For $\frac{1\frac{3}{4}}{32}$, the amount is $\frac{3}{4}$, and so on.) The result is the percentage of loss.

Example. — $\frac{3}{32}$.

Feed-water, 110°; total heat, 1193·45°; sensible steam, 260°.

260° — 110° = 150° heat lost by blowing off.

1193·45° — 110° = 1083·45° total heat.

1083·45° × 2 = 2166·9° + 150° = 2316·9° total heat imparted, and loss by blowing off.

(150° × 100) ÷ 2316·9° = 6·47 per cent. of heat lost by blowing off.

The Barometer.

The Barometer is an instrument used for observing the pressure and elasticity, or variations in density, of the atmosphere. Its essential part is a well formed glass tube, closed at one end, perfectly clear and free from flaws, 33 or 34 inches long, of equal bore, filled with pure mercury, and inverted; the open end being inserted in a cup partly filled with the same metal, so that the mercury in the tube may be supported by atmospheric pressure.

When the air is dry and light, the mercury in the barometer rises; when the air is humid and heavy, it falls. When changes in the weight of the atmosphere take place gradually, they are

imperceptible to human sensation; and if it were not for this instrument, it would be impossible to estimate accurately atmospheric conditions. If, in fine, clear weather, a rain-storm is approaching, the increasing humidity of the atmosphere will be noted by the fall of the barometer long before it will be perceived by ordinary observers. Hence, the condition of the barometer is an indication of not only the weather at the time, but of that which is to follow during the course of several hours. It is in a constant state of variation, governed by the condition of the air. The mercury in the barometer stops falling at 30 inches at sea-level.

TABLE

SHOWING THE WEIGHT OF THE ATMOSPHERE PER SQUARE INCH CORRESPONDING WITH DIFFERENT HEIGHTS OF THE BAROMETER.

Barometer in Inches.	Atmosphere in Pounds.	Barometer in Inches.	Atmosphere in Pounds.	Barometer in Inches.	Atmosphere in Pounds.
28·0	13·72	29·1	14·26	30·1	14·75
28·1	13·77	29·2	14·31	30·2	14·80
28·2	13·82	29·3	14·36	30·3	14·85
28·3	13·87	29·4	14·41	30·4	14·90
28·4	13·92	29·5	14·46	30·5	14·95
28·5	13·97	29·6	14·51	30·6	15·00
28·6	14·02	29·7	14·56	30·7	15·05
28·7	14·07	29·8	14·61	30·8	15·10
28·8	14·12	29·9	14·66	30·9	15·15
28·9	14·17	30·0	14·70	31·	15·19
29·0	14·21				

Thermometers.

The Thermometer is an instrument for measuring variations of heat or temperature. It consists of a bulb and glass stem of uniform bore. A sufficient quantity of mercury having been introduced, it is boiled, to expel the air and moisture, and the tube is then hermetically sealed. The properties of mercury which render it preferable to all other liquids are these: it supports,

before it boils, more heat than any other fluid, and endures a greater cold than would congeal most other liquids.

The Hotwell Thermometer.

The standard points are ascertained by immersing the thermometer in melted ice and in the steam of water boiling under the pressure of 14·7lbs. on the square inch, and marking the positions of the top of the column. The interval between those points is divided into the proper number of degrees,—100 for the Centigrade scale, 180 for Fahrenheit's, and 80 for Reaumur's.

The Uptake Thermometer.

The word "zero" is of Arabic origin, and means empty; hence nothing. Absolute zero is a temperature which is fixed by reasoning, although no opportunity ever occurs for observing it. It is the temperature corresponding to the disappearance of gaseous elasticity; or, in other words, the point where gas would become a solid, as where water becomes ice. This temperature is called zero in reference to all the gases, and the positions of the absolute zero on the ordinary scales would be

Reaumur's scale	219·2 below 0°
Centigrade	244 "
Fahrenheit	461·22 "

Rules *for Comparing Degrees of Temperature Indicated by Different Thermometers.* 1. *Multiply* degrees of Centigrade by 9 and *divide* by 5; or *multiply* degrees of Reaumur by 9 and *divide* by 4. Add 32 to the quotient in either case, and the sum is degrees Fahrenheit. 2. From degrees of Fahrenheit *subtract* 32; *multiply* the remainder by 5, and divide by 9 for degrees Centigrade; or *multiply* by 4, and *divide* by 9 for degrees Reaumur. The abbreviation for Fahrenheit is "Fah."; for degree, °.

Marine Steam-Engine Register.

This instrument is designed for application to marine steam-engines. It consists of a circular box faced with a dial, in which are cut, side by side, six or more slots, through which may be seen the numbers representing the revolutions of the engine. This dial is called the "counter" or "register," which is worked by an attachment to any suitable part of the engine, from which a vibratory motion may be communicated to an arm attached to a central horizontal shaft placed parallel to the dial, into the ends of which

is fixed a frame carrying a small shaft, parallel to the former, to which six arms are attached in such a way that the right arm may fall without the others, but cannot rise without carrying all the rest.

This framework, with the pall-shaft, etc., by the motion of the arm attached to the engine, describes an arc of 36°, or $\frac{1}{10}$ of a circle. The ends of the palls, respectively, rest on and slide over 6 cylinders placed side by side on the central shaft, all of which move in the same direction, and are numbered from right to left. On the right-hand edge of each cylinder are cut 10 slots, and on the left hand only one slot, which are of such a

size as to admit the end of one of the palls. Then, on the back motion of the framework, etc., the pall is carried back until it drops in, when the forward motion carries with it the cylinder so locked.

In the spaces between the laps, in each cylinder, and opposite to one of the slots in the dial face, the numbers 1, 2, 3, etc., to 0, are engraved at equal distances around the circumference. The palls are placed one over each of the slots, so that the pall can fall into the inner cylinder only when the slot in the outer one comes directly under it. As this occurs only once in a whole revolution, and as the motion of the palls is only through one-tenth of a circle, it follows that cylinder No. 2 can only be moved through one-tenth of its circumference after cylinder No. 1 has moved a whole revolution, or ten times that space, and so on. Thus, if the figures on No. 1 represent units, those on No. 2 will be tens, on No. 3, hundreds, etc. It will be observed that every revolution of the engine insures one-tenth of cylinder No. 1 to move round, inasmuch as the ten slots in its right-hand edge are not covered by any other cylinder, as is the case with the others.

Rule *for Finding the Number of Revolutions the Engine has made during the Voyage.*

Subtract the number at which the counter stood at the beginning of the voyage from that which is indicated at the end of it; the remainder will be the number of revolutions made during the voyage.

To Reduce the Time the Counter has been Working into Minutes.

Multiply the days by 24,* the product will be the hours; *multiply* this by 60,† the result will be the minutes during which the counter has been working, or *divide* the number of revolutions by the minutes the counter has been working; the quotient will be the average number of revolutions made by the engine per minute.

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* 24 hours being equal to one day. † As 60 minutes = 1 hour.

Spring-, Mercury-, Syphon-, and Vacuum-Gauges.*

Figure I shows an inside view of the Lane spring steam-gauge. As may be observed, it consists of a hollow brass tube, a lever, connecting-link, sector, pinion, and pointer. Its operation is as follows: Pressure is exerted in the tubes, *A*, *A*, through the nipple, *B*, the effect of which is to elongate or straighten it. The consequence is, that the link, *C*, draws the lever, *E*, and the sector, *F*, which moves the pinion, which is not shown, but which carries the pointer, *G*.

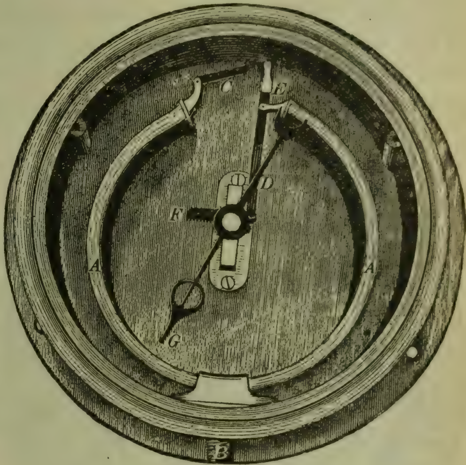


Fig. 1.—Inside View.

The higher the pressure, the more the tubes will be expanded or elongated, and the higher the pointer will be carried up. As the pressure decreases, the tubes have a tendency to contract, and the pointer again assumes its natural position at zero.

Fig. 2 (page 370) **represents the Bourdon** spring steam-gauge. It consists, as in the case of the Lane, of a hollow metal tube, connecting-link, sector, pinion, coil-spring, and hand or pointer. As will be seen, though the mechanism is reversed, the principle is the same as in the Lane gauge. The pressure exerted in the hollow tube, *G*, has a tendency to expand or elongate it; the result of which is, that the link, *H*, draws the sector, *J*, (which swings on the stud *I*) to the right, the upper end of which turns the pinion, *K*, which carries the pointer to the right also. A coil-spring is

* See page 658.

attached to the stud, which carries the pointer to assist in bringing it back to a state of rest, as the pressure decreases.

The advantages of spring-gauges are, that they are light, cheap, and simple, and are not affected by jar or jolting; their disadvantages are, liability to corrode, and the spring losing its tension; they require to be tested and corrected at least once a year. When steam-gauges of any kind are set up, the end of the pipe next the gauge should invariably be filled with cold water. The steam should never be allowed to act directly on a steam-gauge when located in cold situations, where they are liable to freeze. The valve on the boiler should be closed, and the drip attached to the gauge opened, in order to allow the water to run out. The drip on the gauge should be closed before the steam is turned in from the boiler, in order that sufficient steam may be condensed in the pipe to furnish the quantity of water necessary to keep the steam from striking the gauge.

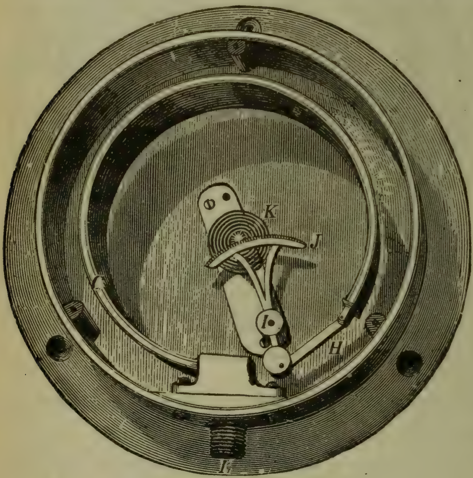


Fig. 2.—Inside View.

The spring-gauge can also be used as a vacuum-gauge, by reversing the application of the pressure, which has a contrary effect on the tube. For instance, as exhaustion takes place in the tube, its power of resisting the pressure of the surrounding atmosphere, which acts upon it, varies also, and it consequently again coils under that pressure in regular ratio with its variation, and indicates the degree of vacuum in the condenser.

A siphon-gauge is a bent tube, inverted, and partially filled with mercury. The orifice of the short leg is connected with the boiler, and the long leg is open to the atmosphere. The steam pressing upon the mercury in the short leg with greater force than the pressure of the atmosphere, causes the mercury in the other leg to rise, and indicates the excess of pressure above the atmosphere. To the amount shown by the gauge must be added the pressure of the atmosphere. Thus, if a siphon-gauge shows 15 lbs. pressure, the boiler-pressure is 30 lbs.

A mercurial gauge, for high-pressure steam-engines, consists of a glass tube open at the lower end, and closed at the top, containing air in its ordinary state. Its lower end is placed in a cistern of mercury. When the cock is opened, the steam passes through, forcing the mercury up the glass tube, thereby compressing the air in the tube above the mercury. When the air is compressed to one-half its original space, the pressure is doubled; to one-third, it is trebled; to one-fourth, it is quadrupled, etc.

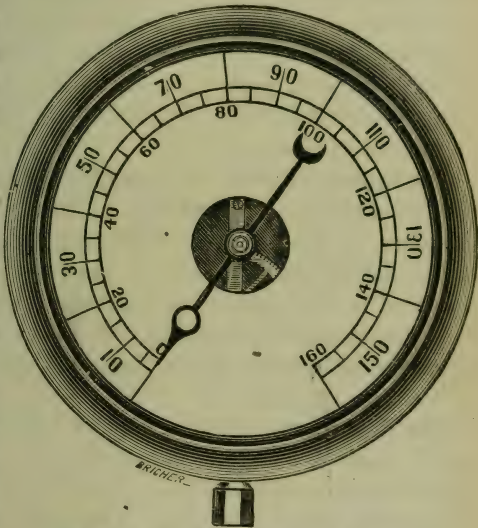


Fig. 3. – The Spring Steam-Gauge.

A barometer-gauge is a tube of glass, more than 30 inches long, closed at one end, and filled with mercury, then inverted so that the lower or open end will be immersed in a cistern of mercury, when the mercury in the tube will sink, rising in the basin until its weight balances the pressure of the atmosphere, which, by its

elasticity, is endeavoring to force the mercury up the tube. The mercury in the tube will be found to stand about 30 inches higher than the level in the basin, varying slightly, according to the state of the atmosphere.

The scale of a barometer-gauge may be explained as follows: As 30 inches of mercury press down with the same force as the atmosphere, say 15 lbs. per square inch, two inches of mercury correspond to one pound of pressure, and a scale of inches measured from the mercury in the cup upwards must be fixed near the glass tube. As the vacuum, while the engine is working, may be supposed to be good, the scale need only be marked to a few inches below 30 inches, every fall of two denoting one pound of pressure in the condenser.

The sources of error, in estimating the vacuum by this gauge, arise from the following two facts: That the pressure of the atmosphere, or the mercury in the cup, is liable to change. That the gradations on the scale are marked, on the supposition that the level of the mercury is stationary; because it is from this level that the scale commences. Therefore a fixed scale must be erroneous, on account of the sinking of the mercury in the cup as it rises in the tube.

The first source of error may be corrected by observing the actual height of a weather barometer, and subtracting it from the height as shown by the gauge. This will be correct, if a tube of a standard diameter is used. This error may be corrected by a short gauge, similar to what a weather barometer would be if it were enclosed in a space, communicating with the condenser. In that event, before a vacuum is created, the mercury would stand as high in the glass tube as in the weather barometer. On creating a vacuum, thus taking off the pressure from the mercury in the cistern, the mercury would fall in the tube. In this instrument, the less the height of the mercury the better the vacuum.

The second source of error may be obviated by having a movable instead of a fixed scale, so that its lower end might always be kept in contact with the surface of the mercury in the cup.

A siphon-gauge, such as has been spoken of, may be used as a vacuum-gauge. When so used, it is necessary to connect the long leg with the condenser, placing a stick in the short leg. In this case the scale would require to be graduated directly contrary to that for steam. The state of the atmosphere will affect the gauge. The pressure in the steam-boiler may be ascertained by the temperature, by the safety-valve, or by the steam-gauge.

The Mariner's Compass.

The object of the mariner's compass is to enable travellers to steer their course with certainty from one location to another. The needle is understood to point to the north, and the other points, east, west, etc., are easily found. In certain parts of the world, however, the needle does not point to the north, but is drawn to the right or left of true north. This is called the variation of the compass, and must be known accurately by the navigator, in order to correct and steer the right course. For instance, in crossing the Atlantic Ocean, the variation of the compass amounts in sailing vessels to $2\frac{1}{2}$ or $2\frac{3}{4}$ points westerly, and the course steered must be corrected accordingly. If a due east course is desired, the vessel must be steered $2\frac{1}{2}$ or $2\frac{3}{4}$ points south.

Off the Cape of Good Hope, the variation of the compass in ships bound to India or Australia is $2\frac{3}{4}$ points easterly, and, in order to make a due east course, it is necessary to steer $2\frac{3}{4}$ to the north, or left of her course; while towards the equator there is hardly any perceptible variation of the compass at all. The best means of finding out how much the compass varies in different parts of the world is by observations of the sun taken with the compass, and the difference between the true and magnetic compass is the variation, which must be applied as a correction to the course steered. In iron ships or steamers, the deviation must be considered as well as the variation. This is due to the local attraction caused by the iron, and must be carefully understood before steamers or iron ships go to sea. Before a vessel proceeds on her first voyage, the compass must be carefully swung and magnets fixed to the deck.

TABLE

OF RHUMBS, OR POINTS OF THE COMPASS.

Points.	Angles. ° ′ ″	NORTH.	NORTH.	SOUTH.	SOUTH.
¼	2 48 45	N ¼ E	N ¼ W	S ¼ E	S ¼ W
½	5 37 30	N ½ E	N ½ W	S ½ E	S ½ W
¾	8 26 15	N ¾ E	N ¾ W	S ¾ E	S ¾ W
1	**11 15 0**	N by E	N by W	S by E	S by W
1¼	14 3 45	N by E ¼ E	N by W ¼ W	S by E ¼ E	S by W ¼ W
1½	16 52 30	N by E ½ E	N by W ½ W	S by E ½ E	S by W ½ W
1¾	19 41 15	N by E ¾ E	N by W ¾ W	S by E ¾ E	S by W ¾ W
2	**22 30 0**	NNE	NNW	SSE	SSW
2¼	25 18 45	NNE ¼ E	NNW ¼ W	SSE ¼ E	SSW ¼ W
2½	28 7 30	NNE ½ E	NNW ½ W	SSE ½ E	SSW ½ W
2¾	30 56 15	NNE ¾ E	NNW ¾ W	SSE ¾ E	SSW ¾ W
3	**33 45 0**	NE by N	NW by N	SE by S	SW by S
3¼	36 33 45	NE ¾ N	NW ¾ N	SE ¾ S	SW ¾ S
3½	39 22 30	NE ½ N	NW ½ N	SE ½ S	SW ½ S
3¾	42 11 15	NE ¼ N	NW ¼ N	SE ¼ S	SW ¼ S
4	**45 0 0**	NE	NW	SE	SW
4¼	47 48 45	NE ¼ E	NW ¼ W	SE ¼ E	SW ¼ W
4½	50 37 30	NE ½ E	NW ½ W	SE ½ E	SW ½ W
4¾	53 26 15	NE ¾ E	NW ¾ W	SE ¾ E	SW ¾ W
5	**56 15 0**	NE by E	NW by W	SE by E	SW by W
5¼	59 3 45	ENE ¾ N	WNW ¾ N	ESE ¾ S	WSW ¾ S
5½	61 52 30	ENE ½ N	WNW ½ N	ESE ½ S	WSW ½ S
5¾	64 41 15	ENE ¼ N	WNW ¼ N	ESE ¼ S	WSW ¼ S
6	**67 30 0**	ENE	WNW	ESE	WSW
6¼	70 18 45	ENE ¼ E	WNW ¼ W	ESE ¼ E	WSW ¼ W
6½	73 7 30	ENE ½ E	WNW ½ W	ESE ½ E	WSW ½ W
6¾	75 56 15	ENE ¾ E	WNW ¾ W	ESE ¾ E	WSW ¾ W
7	**78 45 0**	E by N	W by N	E by S	W by S
7¼	81 33 45	E ¾ N	W ¾ N	E ¾ S	W ¾ S
7½	84 22 30	E ½ N	W ½ N	E ½ S	W ½ S
7¾	87 11 15	E ¼ N	W ¼ N	E ¼ S	W ¼ S
8	**90 0 0**	EAST.	WEST.	EAST.	WEST.

TABLE

SHOWING THE MAGNITUDES AND VELOCITIES OF THE PLANETS.

PLANETS.	DIAMETERS.	RELATIVE MAGNITUDES AS COMPARED WITH THE EARTH.	ORBITAL VELOCITY.	DISTANCE FROM SUN.
	Miles.		Miles per Sec.	Miles.
Mercury . .	3,089	$\frac{1}{17}$ of magnitude of the earth. . .	29	35,392,000
Venus . .	7,100	Equal " " "	22	66,134,000
Earth . . .	8,000		19	93,321,000
Mars . . .	4,115	$\frac{1}{7}$ of " " "	16	139,311,000
Asteroids . .	30 to 500	$\frac{1}{266}$ to $\frac{1}{160}$ " " "	12	Various.
Jupiter . . .	89,000	1,400-fold greater than the earth.	8	475,692,000
Saturn . .	70,000	1,000 " " "	5	872,137,000
Uranus . .	34,000	83 " " "	4	1,753,869,000
Neptune . .	37,000	105 " " "	$2\frac{1}{2}$	2,745,998,000
The Sun .	887,000	1,300,000 " " "	$4\frac{1}{4}$	Central Orb.
	The Sun is more than seven hundred-fold greater than all the planets combined.			

Technical Terms and Definitions Used in Navigation.

Apparent altitude. — The apparent altitude is the observed altitude, corrected for the indicated error of the instrument, and dip of the horizon.

Meridian altitude. — The meridian altitude is the highest altitude a celestial object attains on the meridian of the observer.

Observed altitude. — The observed altitude is the altitude of a celestial object above the horizon measured by a sextant or quadrant.

True altitude.— The true altitude is the apparent altitude corrected for refraction and parallax.

Amplitude. — The amplitude is the arch of the horizon contained between the centre of the celestial object, when rising or setting, and the east or west points of the horizon, measured from the east when rising, and from the west when setting.

Azimuth. — An azimuth is the angle at the zenith contained between the vertical circle passing through the centre of the celestial object, and of the meridian of the place.

Course. — The course is the direction steered by compass.

Magnetic course. — The magnetic course is the compass course corrected for deviation of the compass.

True course. — The true course is the compass course corrected for variation and deviation of the compass.

Course made good. — The course made good is the compass course corrected for deviation, variation, leeway, and set of the current, and is the ship's real track on the ocean.

Variation of the compass. — The variation of the compass is the angle between the true north and the magnetic north. There are only few places where the needle points exactly to the true north. When it points to the eastward of the true north, it is easterly variation; but when the north point of the needle is attracted to the westward of north, it is called westerly variation.

Deviation of the compass. — The deviation of the compass is the angle between the compass north and the magnetic north, and is produced by the local attraction of the ship's iron on her compasses.

Declination.— The declination is the distance a celestial object is north or south of the equinoctial, measured on a meridian.

Error of the compass. — The error of the compass is the variation and deviation combined.

Dead reckoning. — The dead reckoning is the method of ascertaining the ship's position by the courses steered and distance sailed, as shown in the following pages under the head of the Day's Work. This is liable to many errors, such as bad steering, unknown currents, improper allowances made for distance run, and often fails to give the ship's true position.

Departure. —The departure is the *distance in miles* made good by a ship, east or west; when a ship sails due north or south, she makes no departure.

Taking a departure.—When bearings are taken of some headland or other known object, before a ship leaves the land, it is called *taking a Departure.*

Distance.—Distance is the distance between two places or positions, or the distance sailed by a ship on a certain course, measured in nautical miles.

Polar distance.—The polar distance is the distance of a ce-

lestial object from the elevated pole, and is found by subtracting the declination of the object from 90°, when the latitude and the declination are of the same name, but by adding the declination to 90°, when they are of contrary names.

Ecliptic. — The ecliptic is the apparent annual path of the sun in the heavens.

Equator. — The equator is a great circle passing round the earth, 90 degrees from the poles, and dividing it into two equal parts or hemispheres, called the Northern and Southern Hemispheres. At all places on the Equator, the sun rises and sets at six o'clock all the year round.

Visible horizon. — The visible horizon is the circle that bounds the observer's view at sea, where sky and water appear to meet.

Dip of the horizon. — The dip of the horizon is the angle between the true and visible horizon, and is a correction which must be subtracted from all altitudes.

Hour angle of a celestial object. — The hour angle of a celestial object is the angle at the pole between the meridian of the observer and that of the celestial object.

Latitude.—Latitude is distance north or south from the Equator, measured in degrees, minutes, and seconds on a meridian; a place or position is in north or south latitude, according as it is north or south of the Equator; a degree of latitude is 60 nautical miles of 6082 feet.

Parallels. — Parallels of Latitude are small circles parallel to the Equator, running round the earth east and west. Two places situated on one of these circles are said to be in the same parallel of latitude.

Difference. — Difference of Latitude is the distance a ship

makes good in a north or south direction. When two places or positions are on the same side of the Equator, that is, in north or south latitude, their difference of latitude is found by *subtracting* the lesser latitude from the greater; when two places or positions are on the opposite sides of the Equator, that is, when one is in north latitude, and the other in south latitude, their difference of latitude is found by *adding* the latitudes together.

Leeway. — The leeway is the angle between the ship's true course and her path through the water; starboard tack allows leeway to the left hand; port tack allows it to the right hand.

Longitude.— Longitude is the degrees, minutes, and seconds a place or position is east or west of the first meridian, measured on the Equator. Most nations adopt the Meridian of Greenwich observatory in England as the first meridian. Thus the longitude of a place or position is called east or west of the Meridian of Greenwich, reckoned up to 180 degrees, which is the opposite meridian to Greenwich, or one-half of the circumference of the earth. Longitude is also reckoned by time,— hours, minutes, and seconds,— each hour being equal to 15 degrees of longitude, as the sun, which regulates the time, returns to the same meridian once in every 24 hours. Thus 15 degrees multiplied by 24 hours makes 360 degrees, the entire circumference of the earth.

To reduce longitude into time. — Divide the number of degrees, seconds, and minutes by 15, and the quotient will be the time.

Degrees of Longitude. — The degrees of longitude are of the same length at the Equator as a degree of latitude, viz., 60 nautical miles; but as the meridians contract, and the distance between them decreases gradually the farther you go north or south, until they meet at the poles, it is evident that the space contained in a degree of longitude becomes less the farther north or south the distance travelled. Thus in latitude 60° north or south, 30 miles of departure is equal to a degree of longitude. It will be

seen that if a vessel sails 60 miles east or west in the parallel of 60° north or south, she will make two degrees of longitude; in latitude of 70° north or south, 60 miles is equal to nearly three degrees of longitude.

Difference of longitude. — The difference of longitude is the difference in degrees, minutes, and seconds which one place or position is east or west of another; when two places or positions are on the same side of the Meridian of Greenwich east or west, their difference of longitude is found by subtracting the less from the greater. When they are on opposite sides of the Meridian of Greenwich, that is, one in east longitude and one in west longitude, their difference of longitude is found by adding the two together. When one longitude is east and the other west, and on being added together the sum *exceeds* 180 degrees, it must be subtracted from 360 degrees to get the difference of longitude.

Meridian. — A meridian is a circle passing through both poles, and crossing the Equator at right angles All places situated on this circle are on the same meridian, or in the same longitude north or south of each other.

Parallax. — The parallax is the difference between the altitude of a heavenly body observed on the surface and what it would be if taken at the centre of the earth.

Poles. — The poles are the extremities of the earth's axis; these are 90 degrees north and south of the Equator, and are called the North and South Poles.

Port side. — The term port side is used to designate the left hand side of the ship looking towards the bow.

Refraction. — The refraction is the difference between the real and apparent places of heavenly bodies, as affected by the atmosphere.

Right ascension. — The right ascension is the distance a celestial object is east of the first point of Aries, measured on the equinoctial.

Semi-diameter. — The semi-diameter is half the diameter of the sun or moon. It is given for each day in the Nautical Almanac, and must be applied to all altitudes of the sun or moon to get the true central altitude. If the lower limb is observed, it must be *added;* if the upper limb, *subtracted,* and *vice versâ.*

Starboard side. — The term starboard side is employed to designate the right hand side of a ship looking towards the bow.

Augmentation. — The augmentation of the Moon's semi-diameter is a correction to be added to the semi-diameter, as taken from the Nautical Almanac, on account of the moon being nearer to the observer when above the horizon than when in the horizon.

Tropics. — The Tropics are that portion of the earth situated between 23½° north and 23½° south latitudes.

Civil time.—Civil time is reckoned from midnight to noon, then called A. M.; and from noon to midnight, then called P. M. The civil day commences at midnight; the nautical or sea day commences at noon, twelve hours before the civil day.

Astronomical time.—Astronomical time is reckoned from noon to noon continuously, from 0 hour to 24 hours.

Sidereal time.—Sidereal time is the hour-angle of the first point of Aries, west of the meridian.

Apparent time.—Apparent time is time reckoned by the sun, which is subject to continual variations, and requires correction for astronomical purposes.

Mean time.—Mean time is time regulated by the average or mean, instead of the unequal or apparent, motion of the sun,

and is such as would be shown by the sun if it moved uniformly in the equinoctial.

Equation of time.—The equation of time is the difference between apparent and mean time, is found in the Nautical Almanac for each day, and is used for reducing apparent time to mean time.

Zenith distance.—The zenith distance is the distance a celestial object is from the zenith, or the point overhead.

TABLE

OF THE MILE AS MEASURED BY VARIOUS NATIONS.

The English mile is	1760	yds.	The Swedish and Danish mile is	7341·5	yds.
The Scotch "	1984	"	The Arabian "	2143	"
The Irish "	2240	"	The Roman mile is	1628 or 2025	"
The German "	8106	"	The Werst mile is	1167 or 1337	"
The Dutch and Prussian mile is	6480	"	The Tuscan mile is	1808	"
The Italian mile is	1766	"	The Turkish "	1826	"
The Vienna post mile is	8296	"	The Flemish "	6869	"
The Swiss mile is	9153	"			

The British league, or three times our geographical mile of 60 to a degree, or 2025 yards, is 6075 yards. The Brabant league is 6096 yards. The Danish and Hamburg league is 8244 yards; the German league is 8101 yards; the long German league is 10126 yards; the short German league is 6859 yards; the Portuguese league is 6760 yards; the Spanish league is 7416 yards; the Swedish league is 11700 yards. All of them are parts of a degree, but made before the length of a degree was accurately determined.

Length of Days in Different Countries.

At London, England, and Bremen, Prussia, the longest day has 16½ hours. At Stockholm, in Sweden, the longest day has 18½

hours. At Hamburg in Germany, and Dantzic in Russia, the longest day is 18 hours, and the shortest is 7. At St. Petersburg in Russia, and Tobolsk in Siberia, the longest day has 19 hours, and the shortest, 5½. At Tornea, in Finland, the longest day has 24 hours, and the shortest is a half-hour. At Wardbuys in Norway, the longest day lasts from the 1st of May to the 22d of July without interruption; and at Spitzbergen the longest day is three months and a half. At New York the longest day has 15 hours and 56 minutes; and at Montreal 15½ hours.

TABLE

OF SAILING DISTANCES FROM NEW YORK TO DIFFERENT PARTS OF THE WORLD, IN GEOGRAPHICAL MILES.

To Sandy Hook .	18	miles.
“ Nantucket Light .	211	“
“ Boston . . .	302	“
“ Halifax . . .	666	“
“ Cape Henlopen .	149	“
“ Philadelphia .	252	“
“ Cape Henry .	276	“
“ Baltimore . .	428	“
“ Washington .	434	“
“ Norfolk . . .	306	“
“ Richmond . .	375	“
“ Cape Hatteras .	340	“
“ Charleston . .	621	“
“ Savannah . .	716	“
“ Key West . .	1,484	“
“ Havana . .	1,454	“
“ New Orleans .	2,129	“
“ Vera Cruz . .	2,354	“
“ Liverpool . .	3,084	“
“ London . .	3,225	“

To St. Petersburg,	4,420	miles.
“ Havre . . .	3,148	“
“ San Francisco, via Panama,	5,249	“
“ San Francisco, via Cape Horn . . .	18,850	“
“ Melbourne, via Cape of Good Hope . .	12,895	“
“ Nangasaki, Japan . . .	9,800	“
“ Sandwich Islands, via Panama . .	7,157	“
“ Canton, via Panama .	10,000	“
“ Canton, via Good Hope,	19,500	“

There are 5280 feet in a statute mile.

TABLE

OF LATITUDE AND LONGITUDE OF PLACES.

Places.	Latitude.	Longitude.
	D. M.	D. M.
Quebec	46 49 N.	71 16 W.
Halifax	44 38 "	63 65 "
Portland light	43 36 "	70 12 "
Buffalo	42 53 "	78 55 "
Chicago	42 0 "	87 35 "
Newburyport light	42 48 "	70 49 "
Boston State-House	42 21 "	71 4 "
Nantucket light	41 23 "	70 3 "
Newport	41 29 "	71 19 "
New York	40 42 "	74 0 "
Philadelphia	39 57 "	75 10 "
Cape Henlopen	38 46 "	75 4 "
Cincinnati	39 6 "	84 27 "
St. Louis	38 36 "	89 36 "
Richmond	37 32 "	77 27 "
Washington City	38 53 "	77 3 "
Baltimore	39 18 "	76 37 "
Cape Hatteras	35 14 "	75 30 "
Charleston light	32 42 "	79 54 "
Savannah	32 5 "	81 8 "
Cape Florida	25 41 "	80 5 "
Pensacola	30 24 "	87 10 "
Mobile	30 42 "	87 59 "
New Orleans	29 57 "	90 0 "
San Francisco	37 47 "	122 21 "
Cape Horn	55 59 "	67 16 "
Porto Rico	18 29 "	66 7 "
Cape Hayti	19 46 "	72 11 "
Havana	23 9 "	82 22 "
Vera Cruz	19 12 "	96 9 "
Mexico	19 26 "	99 5 "
Porto Bello	9 34 "	79 40 "
Cape St. Augustine	8 21 S.	34 57 "
Rio Janeiro	22 56 "	43 9 "
Buenos Ayres	34 36 "	58 22 "

From Greenwich.

TABLE—(*Continued.*)

OF LATITUDE AND LONGITUDE OF PLACES.

Places.	Latitude. D. M.	Longitude. D. M.
Cape Horn	55 59 S.	67 16 W.
Valparaiso	33 2 N.	71 41 "
London	51 31 "	0 6 "
Liverpool	53 22 "	2 52 "
Greenwich	51 29 "	....
Dublin	53 23 "	6 20 W.
Paris	48 50 "	2 20 E.
Marseilles	43 18 "	5 22 "
Florence	43 46 "	11 16 "
Rome	41 54 "	12 27 "
Naples	40 50 "	14 16 "
Berlin	52 31 "	13 24 "
Hamburg	53 33 "	9 56 "
Vienna	48 13 "	16 23 "
Constantinople	41 1 "	28 59 "
Stockholm	59 21 "	18 4 "
Copenhagen	55 41 "	12 34 "
St. Petersburg	59 56 "	30 19 "
Madrid	40 25 "	3 42 W.
Gibraltar	36 6 "	5 20 "
Lisbon	38 42 "	9 9 "
Palermo	38 12 "	15 35 "
Pekin	39 54 "	116 28 E.
Canton	23 7 "	113 14 "
Cape of Good Hope	34 22 S.	18 30 "
Sidney, Australia	34 0 "	151 23 "
Jerusalem	31 48 N.	37 20 "

From Greenwich.

TABLE

SHOWING THE TIME AT DIFFERENT PLACES WHEN IT IS 12 O'CLOCK NOON AT NEW YORK.

	Hours.	*Min.*	*Sec.*	
Washington, D. C.	11	47	48	A. M.
San Francisco, Cal.	8	46	13	"
Salt Lake City, Utah	9	27	36	"
Greenwich, Eng.	4	56	0	P. M.
Liverpool, "	4	43	59	"
Paris, France	5	5	21	"

TABLE

OF MILES AND KNOTS, KNOTS AND MILES.

The decimals of miles in this table are repeaters, and when four is used, the last figure should be increased by one.

KNOTS.	MILES.	MILES.	KNOTS.
1	1·1515	1	0·868421
2	2·3030	2	1·736842
3	3·4545	3	2·605263
4	4·6060 *	4	3·473684
5	5·7575	5	4·342105
6	6·9090	6	5·210526
7	8·0606	7	6·078947
8	9·2121	8	6·947368
9	10·3636	9	7·815790
10	11·5151	10	8·684211
11	12·6666	11	9·552632
12	13·8181	12	10·421053
13	14·9696	13	11·289474
14	16·1212	14	12·157895
15	17·2727	15	13·026316
16	18·4242	16	13·894737
17	19·5757	17	14·763158
18	20·7272	18	15·631579
19	21·8787	19	16·500000
20	23·0303	20	17·368420
21	24·1818	21	18·236841
22	25·3333	22	19·105262
23	26·4848	23	19·973683
24	27·6363	24	20·842104
33	38·0000	38	33·000000

* There are 6080 feet in a knot.

Marine Signals.

While it must be admitted that we have made great improvement in the design and construction not only of the hulls of steamships, but also of the machinery and all other appliances con-

nected with their use, as a means of river, lake, and ocean navigation, it is also an authenticated fact, that the number of marine disasters increases, especially as regards steamships, and that each succeeding year shows an increase in the loss of steamships as well as of human life and suffering. While light-houses illuminate almost every coast, yet signals of distress become more numerous and more dark, until the surf, as it were, is hoarse with the cries of drowning men. The questions may naturally be asked, in view of the foregoing facts, Have our ship-builders become more unscrupulous? the weather more changeable? or the sea more dangerous? *Within the last thirty-seven years, fifty-six large ocean steamers have been wrecked, involving a loss of* 4780 *lives and over forty millions of dollars' worth of property, and out of the whole number only two were lost from accidents to machinery.*

There are three classes of marine signals in use as a means of warning the mariner of his proximity to danger, viz., day signals, night signals, and fog signals. They address themselves to the eye and to the ear. Day signals, as a rule, are made with flags, as these furnish the simplest and probably the best medium of communication, whenever objects can be made out, and vessels are beyond hailing distance. Besides the light-house, there are three kinds of night signals used which produce sound, viz., the syren, the whistle, and the bell. The light-house, like the flag, is undoubtedly the most reliable and precise when the air is clear; but it frequently unfortunately happens that the strongest lights, even the most powerful electric lights, are often obscured and rendered invisible by fog. As a result, during heavy fogs by day or night, recourse must be had to instruments which produce sound, such as the syren, the whistle, and the bell.

The theory with regard to their use is, that they are capable of emitting sounds of such intensity as to be heard at a distance sufficient to avert impending danger, providing that the officer of the watch is sufficiently wide awake to hear them. It frequently happens that the first indication that the mariner has of his approach to danger is a dull, muffled sound rising slightly above the roar of

the surf, the noise on board, or the wash of the water. He may be undecided as to the character of the sound, or from whence it arose, and, before giving orders, listens for its repetition, but during all this time the ship is rushing on to danger, or perhaps to destruction. Even if he should fully comprehend the nature of the sound and give orders, they may not be fully and quickly comprehended; the steamer may be sluggish in her movements, the engineer may not be at his post or close to the gear, or he may be drowsy and not fully understand the bells. Any of the foregoing circumstances may arise, and, though trivial in themselves under ordinary conditions, are of vital importance when a steamship, freighted with numerous lives and a valuable cargo, is rushing on to danger. Under such circumstances, courage, self-possession, and that spontaneous knowledge of what to do in moments of extreme peril, are invaluable qualifications in the officer in charge.

A steamship from three to four hundred feet in length, that steers well under full steam or sail, must receive a warning signal at least two miles from it in distance, as it will require a circle of at least 5000 feet, or $\frac{9}{10}$ of a mile, in which to turn such a vessel in smooth water; and it will take from ten to fifteen minutes to head her course directly opposite to the one in which she was steering when the signal was given, when she will be found to be nearly, if not quite, a mile from the line in which she was sailing when the helm was put hard over. It will take from seven to ten minutes to head her course in a direction at right angles to the one in which she was steering. In so doing she will describe a semicircle of at least half a mile, and, under the most favorable circumstances of wind and sea, it will take from five to ten minutes to head her square from danger. If all the surroundings were known, the same vessel might be stopped, backed, and be capable of reversing her motion in a period of five minutes. But it must be understood that the foregoing evolutions must be performed under the most favorable circumstances of sea, wind, weather, and sound, which goes to show that a signal to be efficient must be adapted to each and every one of the foregoing cases.

A proper system of lights and signals is of great importance, as they enable the mariner to shorten his voyage, and thus to facilitate travel and cheapen freights. But what is needed is a system by which the signals might be placed by the side of the ordinary track of vessels, indicating to the mariner that he is right and in a position of safety. Vessels might approach such stations in safety, observe their number, and take a new departure from each, the result of which would be that the most dangerous highways of the sea, and the most intricate channels, might be navigated in the most foggy weather.

Marine Whistle-Signals.

When two steamships or boats are approaching each other from opposite directions, *one puff of the whistle* means keep to the right, *thus,* , which will bring the port, or red, light of each vessel in full view of the other.

When two steamers are approaching each other from opposite directions, *two puffs of the whistle* mean go to the left, *thus,*  which will bring the green, or starboard, lights opposite each other.

When two steamboats are moving in the same direction, one behind the other, and the hindermost one wishes to pass the steamboat ahead, if *one puff of the whistle* is given, she passes ahead on the right side, *thus,* , showing the red, or port, light of the passing boat and the green, or starboard, light of the boat being passed. Under the same circumstances, if *two puffs of the whistle* are given, it means that the hindermost boat is coming up on the left side, in which case the passing boat shows the starboard, or green, light, while the boat being passed shows the port, or red, light.

Three puffs of the whistle is a salute, and four or more a call

for an approaching steamer to slow down, stop, or come alongside, as the case may be.

One long puff of the whistle is usually given when backing out of the dock, and *one short puff* is a call to the deck hand.

Marine Bell-Signals.

Steamboat bell-signals for engineers in the mercantile service are as follows:

When the engine is at rest, on receiving *one bell*, the engineer starts ahead slowly, and continues until he receives *a jingle-bell*, which is a signal to steam at full speed.

When running at full speed, *one bell* means to slow down, after which *one bell* signifies to stop. Under the same circumstances, *two bells* in succession signify stop, *four bells* in succession signify reverse and move backward at full speed.

When the engine is at rest, *two bells* signify to go backward at full speed. If it is desired to go slowly backward, the orders are generally sent down through the speaking-tube, or communicated to the engineer by light or heavy taps on the *gong*.

Light Signals for Ocean Steamships.

When under way, a bright white light is fixed on the foremast, so as to show the *light* ten points on each side of the ship, a *green light* on the starboard side, and a red light on the port side, so constructed as to show the same number of points.

Coasting steamers navigating the bays, lakes, rivers, or inland waters of the United States shall carry a *bright light* at the gaff-end or flag-staff, in addition to the side-lights.

Steam-tugs, when towing, must carry two *bright* mast-head *lights* vertically (one above the other), in addition to their side-lights, so as to distinguish them from other steamships.

Fog-signals.—Steamships, when under way, must use a steam-*whistle* at intervals of not more than one minute. Sailing-ships, when under way, must use a fog-horn every five minutes.

Precautions.—In case of two steamships meeting, in order to avoid the risk of collision, the *helms* of both should be put to *port*, so that each may pass on the port side of the other.

Signals of distress.—By day, the firing of a gun at intervals of about a minute, or the distant signal, consisting of a square flag, having a ball above or a ball below, and by night a gun, rocket, or shell fired at intervals of about a minute, or flames on the ship (as from burning a tar-barrel), etc.

Distant signals.—B, Ask name of ship or signal station in sight; C, Yes; D, No; F, Repeat signal—make it more conspicuous; G, Come nearer.

Railroad Signals.

Red signifies danger, and is a signal to *stop*.

Green signifies caution, and is a signal to go *slowly*.

White signifies safety, and is a signal to go on.

Green and white is a signal to be used to stop trains at *flag-stations*.

Blue is a signal to be used by car-inspectors.

Flags of the proper color must be used by day, and lamps of the proper color at night or in foggy weather.

Red flags or red lanterns must never be used as caution-signals; they always signify danger. *Stop*.

A lantern swung across the track, a flag, hat, or any object waved violently by any person on the track, signifies *danger*, and is a signal to stop.

An exploding-cap or torpedo clamped to the top of the rail is an *extra danger-signal*, to be used, in addition to the regular *signals* at night, in foggy weather, and in cases of accident or emergency, when other signals cannot be distinctly seen or relied on.

The explosion of one of these signals is a warning to *stop* the train immediately; the explosion of two is a warning to check the speed of the train immediately, and look out for the regular danger-signal.

A fusee is an extra caution-signal, to be lighted and thrown on

the track at frequent intervals by the flagman of passenger-trains at night whenever the train is not making schedule time between telegraph-stations.

A train finding a fusee burning upon the track must come to a full *stop*, and not proceed until it is burned out.

Train Signals.

Each train, or engine without a train, while running after sunset, or during the day in foggy weather, must display the white *headlight* in front of the engine, and two *red lights* in the rear of the train or engine, except shifting-engines in yards, which will display two *green lights* instead of *red.*

Each passenger-train while running must have a bell-cord attached to the signal-bell of the engine, passing over or through the entire length, and secured to the rear end of the train.

Each passenger-train while running must display one *green flag* at the rear by day, and two green lights, one at each side of the rear car, at night, as markers, to enable operators and enginemen to know that the whole of the train is attached to the engine.

Each freight-train while running must display two *green flags* by day, and two *green lights* at night, one on each side of the rear car, as markers, to enable operators and trainmen to know that the whole of the train is attached to the engine.

Two green flags by day, and two *green lights* at night, carried in front of an engine, denote that the engine or train is followed by another engine or train running on the same schedule. The engine or train thus signalled will be entitled to the same schedule rights and privileges as the engine or train carrying the signals.

Two white flags by day, and two *white lights* at night, carried in front of an engine, denote that the engine or train is extra. These signals shall always be displayed by all work and extra trains or engines, except when running as a regular train.

A blue flag by day, and a *blue light* by night, placed in the draw-head or on the platform or step of a car, at the end of a car stand-

ing on the main track or sidings, denote that car repairmen are at work underneath the cars. The car or train thus protected shall not be coupled to, or move until the blue signal is removed by the car repairmen.

Enginemen's Signals.

One short blast of the whistle is a signal to apply the brakes — Stop. *Thus*, —.

Two long blasts of the whistle is a signal to throw off the brakes. *Thus*, ——— ———.

Two short blasts of the whistle when running is an answer to signal of conductor to stop at next station. *Thus*, — —.

Three short blasts of the whistle when standing is a signal that the engine or train will back. *Thus*, — — —.

Three short blasts of the whistle when running is a signal to be given by passenger-trains, when carrying signals for a following train, to call the attention of trains they pass to the signals. *Thus*, — — —.

Four long blasts of the whistle is a signal to call in the flagman or signalman. *Thus*, ——— ——— ——— ———.

Four short blasts of the whistle is the engineman's call for signals. *Thus*, — — — —.

Two long followed by two short blasts of the whistle when running is the signal for approaching a road crossing at grade. *Thus*, ——— ——— — —.

Five short blasts of the whistle is a signal to the flagman to go back and protect the rear of the train. *Thus*, — — — — —.

A succession of short blasts of the whistle is an alarm for cattle, and calls the attention of trainmen to danger ahead.

A blast of the whistle of five seconds' duration is a signal for approaching stations, railroad crossings, and draw-bridges.

Conductors' Signals.

BY BELL-CORD.

One tap of the signal-bell when the engine is standing is a notice to start.

Two taps of the signal-bell when the engine is standing is a notice to call in the flagman.

Two taps of the signal-bell when the engine is running is a notice to stop at once.

Three taps of the signal-bell when the engine is standing is a notice to back the train.

Three taps of the signal-bell when the engine is running is a signal to stop at the next station.

Signals by Lamp.

A lamp swung across the track is a signal to stop.

A lamp raised and lowered vertically is a signal to move ahead.

A lamp swung in a circle is a signal to move back.

The Screw-Propeller.

The screw-propeller, so commonly applied to the propulsion of vessels, consists of two, three, or four helical or twisted blades set upon a shaft, or axis, revolving beneath the water at the stern. Experience has shown that no screw-propeller, designed or invented up to the present time, has proved superior to all others for all ships. The best propeller for any vessel is the one best suited for that model, regardless of the number of blades, diameter, or pitch. The principles of screw-propulsion embrace those relating to hydraulics also, so that, in proportioning the screw, the lines of the hull should be considered.

The pitch of the screw is the distance that it would advance in one revolution, if working in a solid, fixed nut; or it is the distance between the threads measured in a line with the shaft. The

pitch of a screw, or the circumference of a paddle-wheel, multiplied by the revolutions, and two figures cut off for decimals, gives the speed in knots per hour.

The term left-handed propeller means a screw with a left-handed thread, and a right-handed one has a right-handed thread. A left-handed propeller, to move the ship ahead, goes from left to right, while a right-handed one turns from right to left, looking from the engine-room towards the stern of the boat.

The force which drives a vessel forward when a screw-propeller is used, is the pressure exerted against the thrust-block. Steam being admitted to the cylinder, causes the piston to move, and the motion being transmitted through the connecting-rod to the crank-pin, crank, and propeller shafts, causes the latter to revolve, by which the pressure it exerts against the water is transmitted to the thrust-block, and the vessel forced forward.

The term "slip of the screw" means the difference between the actual advance of the propeller through the water, and the advance which would be accomplished, if there was no recession of the water produced by the pressure of the propelling surface. A screw of 15 feet, if working in a stationary nut, would advance 15 feet for every revolution; but when it acts in the water, it may only advance 14 feet or less, the difference being caused by the water being pressed back, owing to its inertia being inadequate to resist the moving force. In such cases the slip is said to be 1 foot in 15, or nearly 7 per cent. loss.

Measurement of the screw-propeller.—The surface of a screw-propeller is the same as would be generated by a line revolving around a cylinder, through the axis of which it passes, and at the same time advancing along the axis. To find the area of a propeller-shaft, square the diameter, and multiply by the decimal, ·7854.

The Errickson and Delameter propellers are those most generally used, although the Loper Screw, as it is termed, is frequently employed, but it has now nearly gone out of use.

The thrust-block of a propeller is formed of a series of rings,

generally of brass, with spaces between them, into which an equal number of solid collars fit. The thrust is exerted against the fore and aft faces of these rings and collars.

The stern-tube is a tunnel located in the deadwood of a ship, in which the propeller-shaft revolves. Its outer end is made water-tight by a stuffing-box containing a fibrous packing. Stern-tubes were formerly made of wood, but they are now generally made of boiler-plate.

The Paddle-Wheel.

The advantages of the paddle-wheel as a motive-power depend on the amount of the immersion. When the water approaches the centre or reaches above it, it is obvious that great waste of power will ensue. It is quite as obvious that the greater the diameter of the wheel the greater the leverage, and the greater is the effect obtained.

The slip of the paddle is caused by the recession of the water from the buckets, or it is a retrograde motion given to the water in a line parallel to the direction in which the ship is moving. The slip of the wheel is the difference between the speed of the ship and that of the wheel. The amount of slip is determined by finding the speed of the ship in feet per hour and subtracting it from the speed of the wheel at the centre of pressure, or centre of action, in feet per hour.

The centre of action is that point in a wheel in which the effect would not be altered if the whole action of the water were concentrated. The centre of action may be thus determined: Lay down the wheel to a certain scale and line off the dip on it. Take $\frac{1}{3}$ of the breadth of the totally-immersed paddles and $\frac{1}{3}$ of the depth of those partially immersed, and add them; their sum divided by the number of paddles partially or totally in the water will give the distance of the centre of action from the edge of the floats. This distance subtracted from the radius of the wheel, and multiplied by 2, will give the diameter of the centre of action of the wheel. The dip of the wheel is 69 inches, and, at that dip, there are 7 full-immersed floats and one immersed

15 inches, making 8 floats; depth of bucket, 21 inches. Find the speed of the circumference at the centre of action in the paddles in feet per hour; diameter of wheel, 36 feet; revolutions, 15 per minute. First take centre of action at $\frac{1}{3}$ the mean depth of the immersed paddles, then $\frac{1}{3}$ of $21 = 7$, which, multiplied by 7, $= 49$; $\frac{1}{3}$ of $15 = 5$; $5 \times 1 = 5$; $49 + 5 = 54$, which divided by 8, the number of buckets, $= 6{\cdot}75$, which is the distance of the centre of action from the outer edge of the floats. Then $6{\cdot}75 \times 2 = 13{\cdot}5$ inches; 36 feet $= 432$ inches; and $432 - 13{\cdot}5 = 418{\cdot}5 \div 12 = 34{\cdot}883$, which is the diameter of the centre of action of the paddles, and this $\times\ 3{\cdot}1416 = 109{\cdot}5884$, which is the circumference of their centre of action. Then $109{\cdot}5884$ feet $\times$ 15 revolutions $\times$ 60 minutes in an hour $= 98{,}629$ ft., the speed of circumference at the centre of action of the paddles in feet-power as required.

The speed of the ship being $12\frac{1}{4}$ miles per hour, the percentage of slip of the above wheel may be calculated thus: 6082·66 ft. is a nautical mile, which multiplied by $12\frac{1}{4} = 74{,}512{\cdot}58 =$ speed of ship in feet per hour. Then the speed of the centre of circumference of action in feet per hour is $98{,}629 - 74{,}512{\cdot}58$ speed of ship $= 24{,}116{\cdot}42$, slip of centre of action of paddles in feet per hour. The percentage of slip would then be $98{,}629 : 24{,}116{\cdot}42 :: 100 : 24{\cdot}44$, which is the percentage of slip.

Loss from oblique action is the loss in the common radial wheel occasioned by the floats striking the water at an angle. Oblique action consists of a vertical depression and lifting of the water by the entering and emerging floats. This loss is calculated by the square of the sines of the angles at which the buckets strike or enter the water. This may be explained as follows: When a body strikes obliquely on a plane, the force of impact with any given velocity varies according to the sines of the angle of incidence, and therefore the force with which the particles of water strike against a board will vary according to the sines of the angles at which they strike. This force is gradually growing less as the board is turned on its edge. The number of particles striking the board also vary according to the sines of the angles of inci-

dence, or, in other words, according to the perpendicular height of the inclined board; so that the resistance, as it varies both with the force with which the particles strike the board and the number of particles which strike it, must vary according to the squares of the sines of the angle of incidence. This loss from oblique action may be obviated by using the feathering wheel, in which each float is hung upon a centre, and so arranged, by a suitable mechanism, as to be always in a vertical position.

The rolling-circle of a wheel is a circle whose circumference multiplied by the number of revolutions of the wheel in a given time equals the speed of the vessel in the same time; or it is a circle at any point in the circumference, which moves with the same velocity as the speed of the ship. The diameter of the rolling-circle of a wheel is found as follows: Divide the speed of the ship in feet per hour by the number of revolutions per hour; the quotient will be the circumference of the rolling-circle; $10 \times 5{\cdot}280 = 52{\cdot}800 \div 600$ (number of revolutions per hour) $= {\cdot}88$, the circumference of the rolling-circle; and ${\cdot}88 \div 3{\cdot}1416 = 28{\cdot}01$, the diameter of a rolling-circle. A disconnecting-paddle engine is one in which the paddle-wheel can be thrown out of gear by sliding back a clutch on the main-shaft. The crank-pin is made fast in the outer shaft, and, if desired to stop one engine, the throttle is shut, and the clutch on the shaft slipped back, which enables either engine to be reversed or stopped independently of the other.

The names of the different paddle-wheels are the cycloidal, the manly, the radial, and the feathering. The two former, though possessing some good features, for certain reasons have never come into very general use, while the latter are almost universally adopted. The feathering wheel is capable of producing more useful effect with less power than the radial wheel, but it has the disadvantages of great weight, extra first cost, as well as great expense of maintenance.

The usual rule for calculating the horse-power of an engine cannot be applied to calculate the actual horse-power available for propelling a vessel, as much of this power is lost by the slip of the wheel and the oblique action of the buckets.

Comparative efficiency *of the screw-propeller and paddle-wheel.*—When a vessel is propelled through the water, she necessarily puts a column of water into motion in the direction of her advance. In the case of paddle-wheels, none of the power expended in producing the current is recovered; but with the screw the case is different, as the effect of the current reduces the number of rotations requisite for the production of any prescribed speed. In short vessels, in consequence of the rubbing surface of the bottom not being sufficient to generate a current, there is little difference between the performances of the paddle and screw; but in long vessels, where the water is more effectually rubbed into motion, the superior efficiency of the screw over any species of side propeller becomes very conspicuous.

In the early trials between paddle- and screw-vessels, the two instruments appeared to be of about equal efficiency. This comparison was made between the screw and radial paddle. But with the feathering paddle the efficiency is found to be greater in consequence of the floats entering and leaving the water edgewise. Feathering wheels have the further advantage for river navigation, that, as the diameter of the wheel may be reduced without giving rise to other difficulties, the speed of the engine may be increased.

In vessels of moderate dimensions, the screw is found to be of about equal efficiency to the radial paddle, and of somewhat inferior efficacy to the feathering paddle; but in vessels of large size, the screw is found to be of considerably greater efficacy than paddles of any kind.

Relation *between the power and speed of steam-vessels.*—When the relation between the pitch of the screw and the speed of the vessel is considered, and also that the pitch must be determined and the screw made before the vessel is tried, it must be obvious how important it is that marine engineers should clearly understand the laws affecting the motion of solid bodies in fluids. A few words will not, therefore, be out of place on this important subject. When a steamboat makes a voyage between port and port, she in effect excavates a canal between those ports, the transverse

section of which corresponds with the immersed midship section of the vessel. It is true this canal is immediately filled up again, but yet this canal is really cut, and the work of the engine is expended in cutting it. After uniform motion has been attained by the vessel, the work of the engine is transferred to the water pushed out of the canal. Now, for similar speeds, the work per mile, or per hour, must be as the immersed midship section of the vessel.

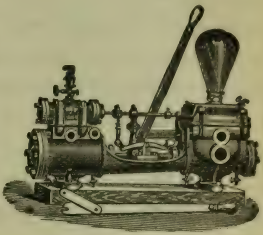
Effect of size on the speed of steam-vessels.—All experiments confirm the theory, that to give one body twice the velocity of another, it will necessarily require the expenditure of four times the amount of energy. Assuming that fluid bodies follow, with respect to motion, the same laws as solids; if two vessels are making voyages, having the same immersed midship section, they will displace similar quantities of water from their course, regardless of speed. The water having motion given to it, the power expended will be in proportion to the square of the velocity given to it.

To find the mean speed of a steam-vessel.— The mean speed of a steam-vessel, from a number of runs over a measured knot where the tidal influence is always varying, may be ascertained as follows: Add the different speeds for the trials, and divide by the number of trials. This will give the approximate mean speed.

The great difficulty with paddle-wheels is to secure a proper immersion. As the ship proceeds on its voyage, and consumes its store of coal, the vessel becomes lighter, and consequently its draught of water decreases. Therefore, supposing a paddle is properly immersed at the commencement of a voyage, it will be partially out of the water at the end. At the commencement of a voyage, the paddle must be too deeply immersed, so that at the middle the proper immersion may be attained, while there will be too little towards the end of the voyage. The paddle-wheel is fast giving place to the screw-propeller, for the reason that it offers greater resistance to the wind in case of storm, thus inducing oscillation of the vessel, besides being more exposed to the shots of an enemy in time of war.

Pumps.

Pumps, of whatever design or construction, or for whatever purpose employed, are simply hydraulic machines attached to one end of a tube, for the purpose of raising, forcing, or transferring water, or other liquids or fluids. The idea entertained by many that water is raised by suction is erroneous, as, properly speaking, there is no such principle as suction. Atmospheric "lift" or "suction" pumps cause the water to raise itself by having its surface relieved of the column of air resting upon it. If, therefore, one end of a pipe or tube be lowered into water, the other end be closed by means of a valve or other device, and the air contained in the pipe be drawn out, it is evident that the surface of the water within the pipe will be relieved of the pressure of the atmosphere. There will then be no resistance offered to the water to prevent its rising in the tube. The water outside of the pipe, still having the pressure of the atmosphere upon its surface, therefore forces water up into the pipe, supplying the place of the excluded air, while the water inside the pipe will rise above the level of that outside of it, proportionally to the extent to which it is relieved of the pressure of the air; so that, if the first stroke of a pump reduce the pressure of the air contained in the pipe from 15 pounds per square inch (which is its normal pressure) to 14 pounds, the water will be forced up the pipe to the distance of about 2¼ feet, since a column of water an inch square, and 2¼ feet high, is equal to about one pound in weight.



It is evident that, upon the reduction of the pressure of the air contained in the pipe from 15 to 14 pounds per square inch, there will be (unless the water ascended the pipe) an unequal pressure upon its surface inside as compared to that outside of the pipe; but, in consequence of the water rising 2¼ feet in the pipe, the pressure on the surface of the water, both inside and outside,

is evenly balanced (taking the level of the outside water to be the natural level of the water inside), as the pressure upon the water exposed to the full atmosphere is 15 pounds upon each square inch of its surface, while that upon the same plane, but within the pipe, will sustain a column of water 2¼ feet high (weighing one pound) and 14 pounds pressure of air, making a total of 15 pounds, which is, therefore, an equilibrium of pressure over the whole surface of the water at its natural level.

If, in consequence of a second stroke of the pump, the air pressure in the pipe is reduced to 13 pounds per inch, the water will rise another 2¼ feet. This rule is uniform, and shows that the rise of a column of water within the pipe is equal in weight to the pressure of the air upon the surface of the water without; hence it is only necessary, to determine the height of a column of water that will weigh 15 pounds per square inch of area at the base, to ascertain how far a suction-pump will cause the water to rise. It must be understood, that the distance varies with the height above sea level, and also with the pressure of the atmosphere. At our level of the sea, the column of water that the atmosphere will support is about 33 feet in height, and a pump will "draw water" (as it is called) this distance; but the force which sends the water into the pump at this height is so diminished as to be almost balanced by its own weight; hence a lifting-pump will deliver water very slowly, drawing it this distance.

To be reliable, the cylinder and piston should be in good order, all the joints perfectly air-tight, a check-valve be placed in the lower end of the suction-pipe; and even then the pumps should be run at a high speed. Pumps will give more satisfactory results when the lift is from 22 to 25 feet. There is hardly any limit to the distance a pump will draw water through a horizontal suction-pipe, provided the pipe is perfectly tight, and everything is so proportioned as not to cause undue friction.

The capacity of any pump may be determined by multiplying the area of the piston in inches by its stroke in inches, giving the number of cubic inches per single stroke; this divided by 231 (the

number of cubic inches in a standard gallon) will give the number of gallons per single stroke; but it must be remembered that all pumps throw less water than their capacity, the deficiency ranging from 20 to 40 per cent., according to the quality of the pump. This loss arises from the lift and fall of the valves, from inaccuracy of fit or leakage, and in many cases from there being too much space between the valves and piston, or plunger. The higher the valves of any pump have to lift to give the necessary opening, the less efficient the pump will be.

The power required to raise a given quantity of water a certain height may be computed by the following rule: Multiply the amount of water in gallons to be raised per minute by 8·35 lbs. (the weight of a gallon of water), and this product by the height, in feet, of the discharge from the point of suction; divide the result by 33,000, which will give the theoretical horse-power required to raise the amount of water to a certain distance. See table on page 522.

The quantity of water which any pump will lift, or discharge, may be estimated by multiplying the area of the piston by the speed; but this rule infers that the pump is fully supplied, and the water thoroughly discharged at every stroke.

Rule for finding the diameter of pump-plunger for any engine.—When the pump-stroke is $\frac{1}{2}$ the stroke of the engine, the diameter of the steam-cylinder multiplied by 0·3 will give the proper diameter of pump-plunger.

Another rule.— When the pump-stroke is $\frac{1}{4}$ of the stroke of the engine, the diameter of the cylinder multiplied by ·42 will give the proper diameter of pump-plunger.

Diameter of pump-plunger should be equal to $\frac{1}{3}$ the diameter of the cylinder when the pump-stroke is $\frac{1}{2}$ the engine-stroke.

Diameter of pump-plunger should be equal to $\frac{1}{6}$ of the diameter of the cylinder when the pump-stroke is $\frac{1}{4}$ the engine-stroke. The velocity of water in pump-passages should not exceed 500 feet per minute. Pump-valves should have an area of $\frac{1}{4}$ the area of the pump.

Feed-pumps for condensing engines.—For condensing engines, the diameter of the pump-plunger should equal 1·11 the diameter

of the steam-cylinder when the pump-stroke is half the engine-stroke, and $\frac{1}{8}$ the diameter of steam-cylinder when the pump-stroke is $\frac{1}{4}$ the stroke of the engine.

Rule to find the diameter of the feed-pump ram.— Multiply the square of the diameter of the cylinder in inches by ·0083. The product is the diameter of the ram in inches. All boiler feed-pumps, when working at ordinary speed, should be capable of discharging one *cubic foot of water* per horse-power per hour.

Rule for finding the necessary quantity of water per minute for any engine.— Multiply the cubic space in the cylinder in inches, to which steam is admitted before being cut off, by twice the number of revolutions per minute, and divide the product by the comparative volume of steam at the pressure used; the quotient will be the cubic inches of water required per minute.

A circulating-pump is used to lift water from the sea and force it through the condenser. Such pumps are not always worked by the main engines, but sometimes are independent or worked by an independent auxiliary engine. See cut on page 352.

Although a pump will require to be in good condition to lift water 33 feet, it will with ease draw water on a level at 1000 feet (providing the pipes are all tight), and force it to any height that the machinery of the pump is capable of bearing.

The reason why pumps do not work is, either that the water-supply is exhausted, the pipes or pistons leak, or the valves prevented from seating. If the valves and connections of a pump are tight and in good order, and it is not located too high above the supply, there is no reason why it should not work.

Pumps become hot from two reasons,—either they are placed too near the boiler, or the pump and check-valves leak, and allow the hot water to escape back from the boiler into the barrel of the pump, which has the effect of expanding the valves and preventing them from doing their work.

A boiler feed-pump, or injector, for any engine should be capable of supplying one cubic foot of water per horse-power per hour. Engines, in general, do not use that amount; in fact, the better

class of automatic cut-off engines will develop a horse-power with a water-consumption of from 25 to 30 lbs.; but it is always best to have the pump or injector sufficiently large, so that, in case the power should be increased, it may be equal to the demand.

An air-chamber is placed on a pump to cushion the water-piston, and relieve the jar that would be induced by the pump-piston striking against a solid column of water; but, to produce the desired effect, it must be perfectly air-tight, otherwise the air will escape. Even when the air-chamber is perfectly air-tight, they require to be frequently refilled, as in fast-running pumps and fire-engines the air becomes condensed. This may be done by stopping the engine or pump, opening a cock or valve that connects with it, and allowing the air to rush in. There is a very general impression among engineers and those having charge of fire-engines, that there is a vacuum in the air-chamber, and the remark is often heard that the pump or engine has lost its vacuum. This is a mistake, as there is no such thing as a vacuum in the air-chamber of a steam-pump or fire-engine. The air-chamber has lost its supply of air either by leakage or condensation. The result is the pump commences to work and labor.

A feed-pump pet-cock, or valve, is a small cock, generally placed on the barrel of the pump above the suction-valve, for the purpose of ascertaining whether the pump is working right or not.

Mud-boxes, strainers, or arresters should be attached to the extreme end of all lift-, suction,- bilge-, or circulating-pumps, for the purpose of arresting any matter that would be liable to choke the pump or prevent the valve from seating.

How to keep pipes and pumps from freezing.—The only certain preventive is the removal of the water from them; consequently, in all cases provision should be made for turning it off during very severe nights. It must be observed, however, that merely shutting off the water is not sufficient; it must all be let out of the pipes. For this purpose a small tap or pet-cock should be placed above the main stop-cock, or the latter should be made with a vent, to allow the water to flow out when it is turned off.

Injectors.

The injector, though simple in design, modest in appearance, and diminutive in size, is, nevertheless, one of the most wonderful, important, and useful machines which the mechanical arts have ever presented to man. It consists of a slender tube, called the steam-tube, through which steam from the boiler passes to another or inner tube, called the receiving-tube. The latter tube conducts a current of water from the pipe into the body of the injector. Opposite the mouth of this second tube, and detached from it, is a third fixed tube, called the delivery-tube. This tube is open at the end facing the water-supply and leading from the injector to the boiler.

Its action is identical to that of the steam-jet, or blower-pipe in the chimney of the locomotive. The principle is, that steam being admitted to the inner tube of the injector, enters the mouth of a combining-tube in the form of a jet, near the top of the inlet water-pipe. If the level of the water be below the injector, the escaping jet of steam, by its superficial action (or friction) upon the air around it, forms a partial vacuum in the combining-tube and inlet-pipe, and the water then rises by virtue of the external pressure of the atmosphere. Once risen to the jet, the water is acted upon by the steam in the same manner as the air has been seized and acted upon in first forming the partial vacuum into which the water rose.

Giffard was the first to make a practical application of the principles embodied in the injector; in fact, when he invented his injector, he may be said to have invented them all. His discovery was, that the motion imparted by a jet of steam to a surrounding column of water was sufficient to force it into the boiler from which the steam was taken, and, indeed, into a boiler working at even a higher pressure. It is not at all extraordinary to see injectors, attached to boilers carrying a pressure of 70 or 80 lbs. per square inch, forcing water into other boilers under a pressure of 250 lbs. per square inch. This extraordinary accumulation of power may be explained as follows: the velocity with which steam

— say at 60 lbs. pressure to the square inch — flows into the atmosphere is about 1700 feet per second. Now suppose that steam is issuing, with the full velocity due to the pressure in the boiler, through a pipe an inch in area, the steam is condensed into water, at the nozzle of the injector, without suffering any change in its velocity. From this cause its bulk will be reduced, say 1000, and therefore its area of cross-section — the velocity being constant —will experience a similar reduction. It will then enter the boiler by an orifice $\frac{1}{1000}$ part of that by which it escaped. Now it will be seen that the total force expended by the steam through the pipe on the area of an inch, in expelling the steam-jet, was concentrated upon the area $\frac{1}{1000}$ of an inch, and therefore was greatly superior to the opposing pressure exerted upon the diminished area.

The invention of the Giffard Injector, like that of the Corliss engine, suggested a numerous progeny. This may be seen from the numerous cuts of that class of machines illustrating this work, but Sellers' Injector is the only one that can be said to be an improvement on Giffard's. All the others are simply modifications of the original Giffard instrument, some few bringing out features which had not been contemplated by Giffard, who considered Wm. Sellers' improvements the only ones that had been made upon his instrument.

Injectors may be divided into three classes — "self-adjusting," "adjustable," and "fixed-nozzle." The *self-adjusting injector* regulates itself to meet all the conditions under which it is intended to work, and, once started, it will work under a variation of steam-pressure of from 10 to 150 lbs. This kind of injector furnishes the most reliable boiler-feeder. The *adjustable injector* is one in which the nozzle can be adjusted to meet the requirements of varying steam-pressure and water-supply. Such injectors are capable of high duty when skilfully managed. The original Giffard represents this type of injector.

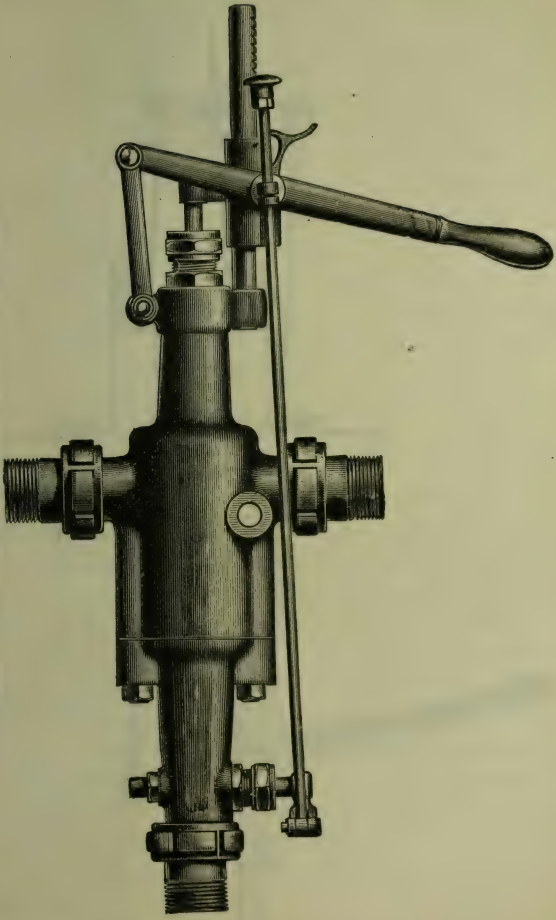
The injector possesses many advantages as a boiler-feeder for furnishing large quantities of water, supplying tanks, etc. Its first cost is moderate, it occupies but little space, and requires no oil,

packing, or repairs. It can be set up almost anywhere and placed either vertically or horizontally; the latter position, however, is preferable. It will act longer, and perform more work even when abused and neglected, than any other device heretofore invented as a boiler-feeder.

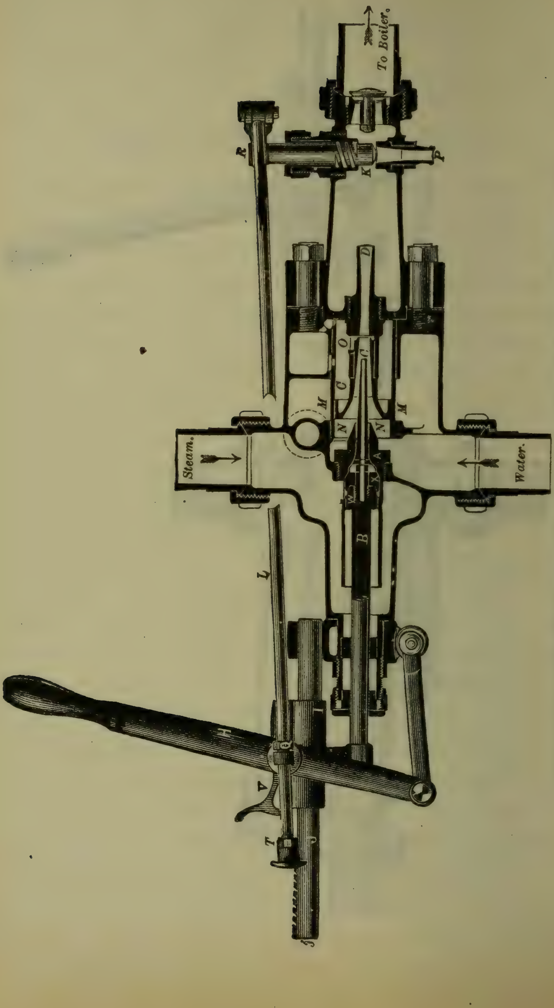
William Sellers & Co.'s Injector.

The cut on page 409 represents Sellers' celebrated lifting injector, so extensively used on locomotives, steamships, tugs, and ferries. It is a self-contained instrument, that is to say, it has both steam- and check-valves; so that it can be connected directly, without any other fittings; although, of course, it is desirable to place another stop-valve in the steam-pipe and a check-valve in the delivery-pipe, so that the injector can be taken to pieces, or disconnected at any time. Another important feature of this injector is, that it is operated by a single handle, and that the waste-valve is only open at the instant of starting.

Its internal mechanism and mode of action may be easily understood by referring to the sectional cut on page 410. *A* is the receiving-tube, which can be closed to the admission of steam by the valve *X*. A hollow spindle passing through the receiving-tube into the combining-tube, is secured to the rod *B*, and the valve *X* is fitted to this spindle in such a way that the latter can be moved a slight distance (until the stop shown in the figure engages with valve *X*) without raising the valve *X* from its seat. A second valve, *W*, secured to the rod *B*, has its seat in the upper side of the valve *X*, so that it can be opened (thus admitting steam to the centre of the spindle) without raising the valve *X* from its seat, if the rod *B* is not drawn out any farther after the stop on the hollow spindle comes in contact with the valve *X*. *D* is the delivery-tube, *O* an overflow opening into space *C K*, the check-valve in delivery-pipe, and *P R* the waste-valve. The upper end of the combining-tube has a piston, *N N*, attached to it, capable of moving freely in a cylindrical portion of the shell, *M M*, and



William Sellers & Co.'s Lifting Injector.



Section of William Sellers & Co.'s Lifting Injector.

the lower end of the combining-tube slides in a cylindrical guide formed in the upper end of the delivery-tube.

The rod *B* **is connected to a cross-head** which is fitted over the guide-rod, *J*, and a lever, *H*, is secured to the cross-head. A rod, *L*, attached to a lever on the top end of the screw waste-valve passes through an eye that is secured to the lever *H*; and stops, *T*, *Q*, control the motion of this rod, so that the waste-valve is closed when the lever *H* has its extreme outward throw, and is opened when the lever is thrown in, so as to close the steam-valve, *X*, while the lever can be moved between the positions of the stops, *P*, *Q*, without affecting the waste-valve. A latch, *V*, is thrown into action with teeth cut in the upper side of the guide-rod, *J*, when the lever *H* is drawn out to its full extent, and then moved back; and this click is raised out of action as soon as it has been moved in far enough to pass the last tooth on the rod *J*. An air-vessel is arranged in the body of the instrument, as shown in the figure, for the purpose of securing a continuous jet when the injector and its connections are exposed to shocks, especially such as occur in the use of the instrument on locomotives.

The manipulation required to start the injector is exceedingly simple,— much more so in practice, indeed, than it can be rendered in description. Moving the lever *H* until contact takes place between valve *X*, and stop on hollow spindle, which can be felt by the hand upon the lever, steam is admitted to the centre of the spindle, and, expanding as it passes into the delivery-tube *D*, and waste-orifice *P*, lifts the water through the supply-pipe into the combining-tube around the hollow spindle, acting after the manner of an ejector or steam-siphon. As soon as solid water issues through the waste-orifice *P*, the handle *H* may be drawn out to its full extent, opening the steam-valve *X* and closing the waste-valve, when the action of the injector will be continuous as long as steam and water are supplied to it.

To regulate the amount of water delivered, move in the lever *H* until the click engages any of the teeth on the rod *J*, thus diminishing the steam-supply, as the water-supply is self-regulat-

ing. If too much water is delivered, some of it will escape through *O* into *C*, and, pressing on the piston *N N*, will move the combining-tube away from the delivery-tube, thus throttling the water-supply; and if sufficient water is not admitted, a partial vacuum will be formed in *C*, and the unbalanced pressure on the upper side of the piston, *N N*, will move the combining-tube towards the delivery-tube, thus enlarging the orifice for the admission of water. The injector, once started, will continue to work without any further adjustment, delivering all its water to the boiler, the waste-valve being kept shut. By placing the hand on the starting-lever, it is easy to tell whether or not the injector is working; and if desired, the waste-valve can be opened momentarily by pushing the rod *L*, a knob on the end being provided for the purpose.

TABLE

SHOWING STEAM-PRESSURE REQUIRED TO LIFT AND DELIVER WATER WITH SELLERS' FIXED-NOZZLE LIFTING INJECTOR.

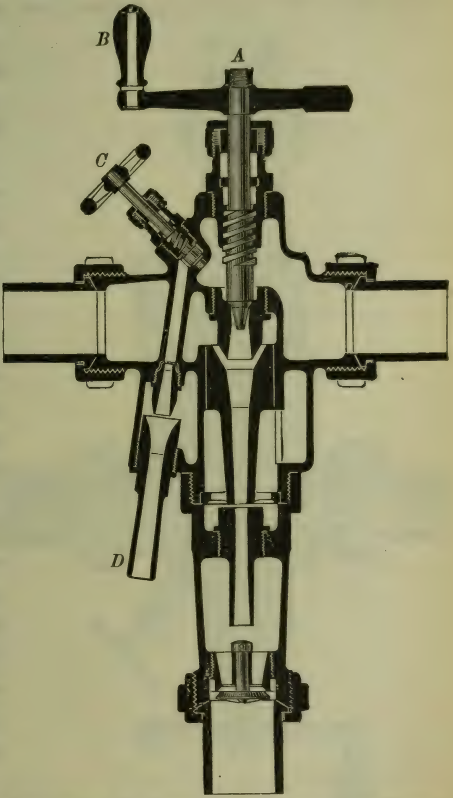
Height Water is Lifted.		Steam-Pressure required to lift and deliver Water.	Height Water is Lifted.		Steam-Pressure required to lift and deliver Water.
Feet.	Inches.	Lbs. per Sq. In.	Feet.	Inches.	Lbs. per Sq. In.
3	0	25	21	3	52
5	0	30	22	10	60
11	6	40	22	10	70
15	0	49	22	10	100

Sellers' Non-Adjusting Fixed-Nozzle Injector with Lifting Attachment, for Stationary Boilers.

The cut on page 413 represents Sellers' Non-Adjustable Injector with fixed-nozzle and lifting attachment. As will be observed, a steam-ejector or siphon is attached to the side of this instrument, which draws the water, when lifted by the admission of the steam, through the combining-tube, and discharges it through the orifice of the lifting attachment, through which, also,

the waste water or overflow escapes. This injector has a check-valve connected to it, also a steam stop-valve, which can be opened wide by half a revolution of the lever on the stem. In connecting the injector, since it has fixed nozzles, a water-supply valve must be provided, and, as already remarked, a second check-valve in the delivery-pipe and another steam-stop valve are desirable.

In starting this injector, steam is first admitted to the lifting-nozzle, the water-supply valve being adjusted so as to deliver about the maximum amount of water corresponding to the steam-pressure; and as soon as solid water issues from the lifting-nozzle, the steam-valve is to be opened slightly until the jet is established, when the full steam-pressure is to be admitted, and the valve that admits steam to the lifting-nozzle is to be closed.

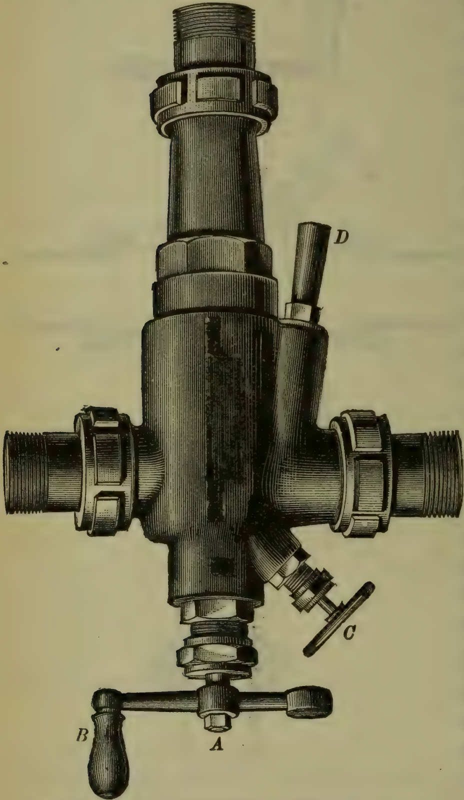


Section of Sellers' Non-Adjustable Fixed-Nozzle Lifting Injector.

Some little dexterity is required to start the injector for a maximum lift, but the manipulation is readily acquired, while for all ordinary lifts no special care is required. As the velocity of steam escaping from an orifice varies greatly with

the pressure, other things being equal, the lifting-nozzle must have proportions depending on the minimum steam-pressure to be employed, since it can readily be adapted to higher pressures by partially closing the steam-admission valve.

Directions for operating Sellers' non-adjustable fixed-nozzle injector, with lifting attachment.—*First*, close the steam-spindle, *A*, by means of the handle, *B*. *Second*, open the lifting-jet by backing the wheel, *C*, one-quarter turn. *Third*, when the water escapes at the overflow, *D*, run out the spindle, *A*, by backing it quickly; then close the lifting-jet, *C*, as the injector will then be feeding the boiler, and the water-supply may be regulated by means of an ordinary globe-valve placed between the injector and the water source. If this valve is set to admit the required quantity of water, there will be no drip from the overflow. When required, a special regulating valve, which requires but one turn, and which indicates the required opening, is attached to the injector, so that those having it in charge may de-



Sellers' Non-Adjustable Fixed-Nozzle Lifting Injector.

termine the actual amount of opening by a glance at the hand-wheel on the valve-spindle.

Duty of Sellers' injectors, or the foot-pounds of useful work performed by the consumption of 100 lbs. of coal in the boiler supplying steam to the injector, may be of interest. When the evaporation of the boiler is known, this duty can readily be computed from the data obtained in connection with the maximum delivery of the injector. This can be illustrated by an example. Assuming the boiler evaporation at 9 lbs. of steam per lb. of coal, a result which, though rather above the average, is occasionally exceeded in good practice. Using the data recorded in the table on page 416 for the maximum delivery at a steam-pressure of 130 lbs. per square inch, it appears that $150 - 66 = 84$ units of heat were imparted to each pound of water delivered by the injector, and, the weight of a cubic foot of water at a temperature of 66° Fah. being about 62·3 lbs., that the total weight of water delivered per hour was $161{\cdot}2 \times 62{\cdot}3 = 10{,}042{\cdot}76$ lbs., so that the total amount of heat imparted to the water per hour was $10{,}042{\cdot}76 \times 84 = 843{,}591{\cdot}84$ units.

The total heat above 32° in a pound of dry steam, at a pressure of 130 lbs. per square inch, is 1187·8 units, and the heat remaining in a pound of steam above 32°, after condensation, is $150 - 32 = 118$ units, so that each pound of dry steam imparted $1187{\cdot}8 - 118 = 1069{\cdot}8$ units of heat to the feed-water, and the weight of dry steam required per hour was $\frac{843{,}591{\cdot}84}{1069{\cdot}8} = 788{\cdot}6$ lbs. The height of a column of water equivalent to the pressure against which the water was delivered was $\frac{144 \times 130}{62{\cdot}3} = 300{\cdot}5$ feet, so that the useful work performed per hour was $10{,}042{\cdot}76 \times 300{\cdot}5 = 3{,}017{,}049{\cdot}38$ foot-pounds. The weight of coal required to do this work, on the assumed boiler evaporation, was $\frac{788{\cdot}6}{9} = 87{\cdot}6$ lbs., so that the duty of the injector, per 100 lbs. of coal, was $\frac{3{,}017{,}049{\cdot}38 \times 100}{87{\cdot}6} = 3{,}455{,}536$ foot-pounds.

The term range is frequently used in connection with injectors, and means the difference between the maximum and minimum delivery.

TABLE

SHOWING THE MAXIMUM AND MINIMUM DELIVERY OF SELLERS' SELF-ADJUSTING, 1876, INJECTOR NO. 6; TEMPERATURE OF DELIVERED WATER; PRESSURE AGAINST WHICH INJECTOR DELIVERS WATER, AND HIGHEST TEMPERATURE OF FEED ADMISSIBLE; WATER FLOWING TO INJECTOR UNDER 15 INCHES HEAD; WASTE-VALVES SHUT.

Pressure of Steam Supplied to Injector, and Pressure against which Water is Delivered. *Lbs. per Sq. In.*	Delivery in Cubic Feet per Hour.			Temperature Fahrenheit Degrees.			Pressure of Steam Required to Deliver Water against Pressure in Column 1.	Highest Temperature admissible of Feed-Water, Fahrenheit Degrees.
	Maximum.	Minimum.	Ratio of Minimum to Maximum Delivery.	Feed-Water.	Delivered Water. At Maximum Delivery.	Delivered Water. At Minimum Delivery		
1	2	3	4	5	6	7	8	9
10	75·3	63·6	0·845	66	100	94	3	132
20	82·4	61·2	0·743	66	108	104	9	134
30	94·2	56·5	0·600	66	114	116	16	134
40	100·1	60·0	0·599	66	120	123	22	132
50	108·3	64·7	0·597	66	124	125	27	131
60	116·5	63·6	0·546	66	127	133	34	130
70	124·8	63·6	0·510	67	130	142	40	130
80	133·0	67·1	0·505	66	134	144	46	131
90	141·3	69·5	0·492	67	136	148	52	132
100	147·2	64·7	0·456	66	140	159	58	132
110	153·0	67·1	0·439	67	144	162	63	132
120	156·6	73·0	0·466	67	148	162	69	134
130	161·2	74·2	0·460	66	150	165	75	130
140	166·0	78·9	0·476	66	153	166	81	126
150	170·7	70·6	0·414	66	157	167	88	121

The table of capacities shows the maximum delivery, but the injector can be regulated so as to reduce the amount about 60 per cent.

TABLE

OF CAPACITIES OF SELLERS' INJECTORS.

Size of Injector.	Size of Pipe for Connections.	Pressure of Steam in Pounds.									
		10	20	30	40	50	60	70	80	90	100
		Cubic Feet of Water Discharged per Hour.									
No. 2	½ in.	8·3	·9	9·7	10·4	11·1	11·8	12·5	13·2	13·9	14·6
" 3	¾ "	19·27	21·04	22·81	24·58	26·35	28·12	29·89	31·66	33·43	35·2
" 4	1 "	36·66	39·6	42·74	45·88	49·02	52·16	55·3	58·44	61·58	64·72
" 5	1¼ "	57·58	62·5	67·42	72·34	77·26	82·18	87·1	92·02	96·94	101·86
" 6	1¼ "	83·48	90·6	97·72	104·84	111·97	119·09	126·21	133·33	140·45	147·57
" 7	1½ "	114·03	123·75	133·48	143·2	152·93	162·65	172·38	182·1	191·83	201·55
" 8	2 "	149·2	162·	174·8	187·6	200·4	213·2	226·	238·8	251·6	264·4
" 9	2 "	189·2	205·35	221·51	237·66	253·82	269·97	286·13	302·28	318·44	334·59
" 10	2 "	233·84	253·8	273·76	293·72	313·68	333·64	353·61	373·57	393·53	413·49
" 12	2½ "	337·2	366·	394·8	423·6	452·4	481·2	510·	538·8	567·6	596·4
" 14	2½ "	451·49	491·45	531·41	571·36	611·32	651·27	691·23	731·18	771·14	811·09
" 16	3 "	600·32	651·6	702·88	784·16	805·44	856·72	908·	959·28	1010·56	1061·84
" 18	3 "	760·07	825·	889·93	954·86	1019·78	1084·71	1149·64	1214·57	1279·5	1344·42
" 20	3½ "	938·84	1019·	1099·16	1179·32	1259·48	1339·64	1419·8	1499·96	1580·12	1660·28

TEMPERATURE OF FEED-WATER.

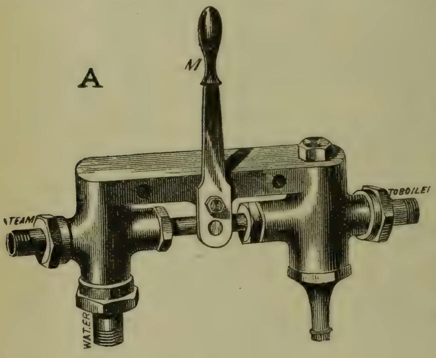
MAXIMUM TEMPERATURE OF FEED-WATER ADMISSIBLE AT DIFFERENT PRESSURES OF STEAM.

Pressure of Steam Pounds per Square Inch.........	10	20	30	40	50	100
Temperature of Feed-water, Fahrenheit.............	148°	138°	130°	124°	120°	110°

The reputation of the Sellers' injectors stands deservedly high for efficiency, reliability, and action. They are adapted to all purposes for which such instruments are employed, such as boiler-feeders for steamships, locomotives, and stationary engines. Every injector is tested at the works by being attached to a steam-boiler before being sent out, and tried under different pressures, which insures entire satisfaction in the working, so that it is sure to meet all the requirements for which it is intended.

Rue's "Little Giant" Injector.

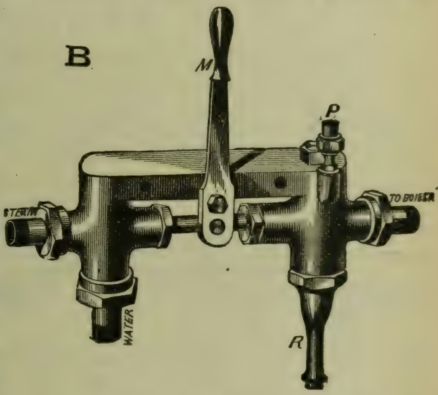
The annexed cut represents Rue's "Little Giant" Injector, class A, which, as a boiler-feeder, has a reputation for simplicity, efficiency, and capacity second to no other in the country. All that is necessary to be done in order to start it, is to turn on the water, and, when it flows from the overflow, turn on the steam, slowly at first, until it reaches the water, then turn on full head, and push the lever *M* slowly, either



forwards or backwards, as seems requisite, until neither steam nor water shows at the overflow. When it is ascertained where the lever must be set, for the steam carried, it can be adjusted before beginning, or left as it is, when steam is shut off. The lever is only used to regulate the proportionate amounts of steam and water. The injector will feed one-half its capacity, by decreasing the amount of steam, and then adjusting the lever. By a little practice, any engineer can adjust it, so as to feed a steady stream of exactly the amount necessary for use.

To set up Class A.— Place the injector in a horizontal position, at any convenient point, so that the pipes will be as short and straight as possible. Place an ordinary globe-valve on the steam-pipe; attach the steam-pipe to the swivel marked steam, and the water-pipe to that marked water; place a valve or stop-cock on the same, as near the injector as practicable. If the water-supply is from a tank, let the fall be as great as possible, but if from a hydrant, or any other source having a pressure which is not regular, as is frequently the case, let the water-pipe be one size larger than the swivel, and attach it to the latter by a reducer. Attach the delivery-pipe to the swivel marked — "to boiler."

The following cut represents the "Little Giant" Lifting Injector, class B, which is used for locomotives and steamships, and will lift water 12 feet at 40 lbs. pressure. This injector, whether used on a locomotive or steamship, should be conveniently located to the engineer. The method of working it is identical with that of class A, except that the steam-jet valve should be first opened, then the water turned on, and, when it appears at the overflow, the main steam supply-valve should be opened gradually until it catches the water, when it may be turned on full head. The steam for them should be taken from the highest point in the boiler, so that it may be dry and elastic. Great care should be taken to see that the water-pipes are all perfectly tight. No washers should be used on the swivels by which the steam- and water-supply pipes are attached to the injector. If there are any floating particles, such as sawdust, shavings, straw, bran, or chaff



in the water, the end of the pipe should be covered with a wire-strainer.

To start the lifting injector, Class B, open the jet-valve until water shows at the overflow in a solid stream; then turn on the steam as before, and, when the water is entering the boiler, shut the jet. Great care should be taken to see that the supply-pipe, through which the water is lifted, is perfectly air-tight, as any leak in the pipe will interfere with the working of the injector. When water is to be lifted by this injector, a small steam-pipe leading from the boiler, and furnished with a valve that opens with a quick motion, is attached to the swivel, *P*, by means of which a steam-jet is thrown into the tube, *R*, and the water lifted.

TABLE

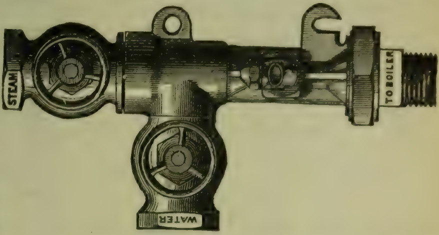
OF CAPACITIES OF RUE'S "LITTLE GIANT" INJECTOR.

Size of Injectors.	Size of Pipe Connections.	Pressure of Steam in Pounds.	Gallons per Hour.	Nominal Horse-Power.
0	¼	90	60	4 to 8
1	⅜	90	90	6 " 12
2	½	90	120	8 " 20
3	¾	90	300	20 " 40
4	1	90	600	40 " 80
5	1¼	90	900	60 " 120
6	1¼	90	1200	80 " 160
7	1½	90	1620	140 " 225
8	2	90	2040	200 " 275
9	2	90	2480	250 " 350
10	2	90	3000	300 " 400
12	2½	90	3600	350 " 500

Friedman's Injector.

The annexed cuts represent the Friedman Injector, which is adapted to a great variety of purposes, such as raising water

or other fluids from tanks, wells, mines, quarries, cellars, docks, the holds of vessels, etc. They have also been successfully employed in breweries, distilleries, chemical works, and sugar refineries, for conveying acids, fluids, or liquids from tank to tank, or from one room to another. In general appearance the injector is of cylindrical form, with three openings, as in all others; one of each for the suction, steam, and delivery. As will be observed, as in all other injectors, instead of one nozzle or cone there are a series in this injector; in this arrangement lies the secret of its capacity and utility.



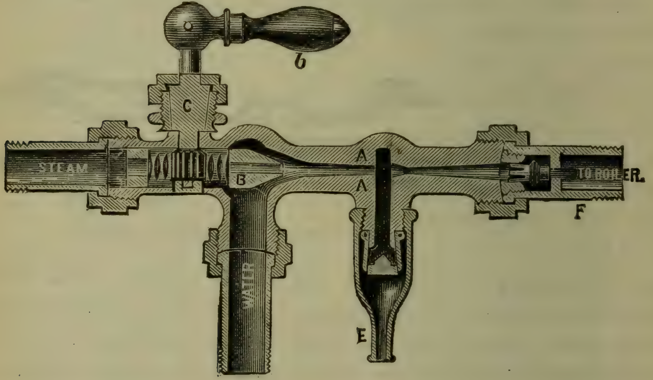
As the steam-jet acts at first only on that portion of the incoming water which is admitted through the first nozzle, or cone, so that only a comparatively small jet of steam is required to move it, this stream, propelled by the force of the steam, gives an impetus to the water entering through the second cone, and that in turn becomes a motor to the next, and so on until the last is reached. The water or liquid accelerated in its passage through these successive nozzles or cones, as well by the force already described as by the vacuum always formed under such conditions, is carried with great velocity through the diverging-pipe into the discharge-pipe, with all the force and rapidity necessary to convey it to its required destination.

Section of Friedman's Injector.

TABLE

OF CAPACITIES OF FRIEDMAN'S INJECTORS.

Size of Injector.	Minimum Inside Diameter of Pipe in Inches.	Delivery per Hour, in Gallons, at a Steam-Pressure of			
		120 lbs.	80 lbs.	50 lbs.	20 lbs.
No. 2	½	90	80	63	39
" 3	¾	220	180	141	90
" 4	1	390	320	243	160
" 5	1¼	630	500	395	250
" 6	1¼	870	720	570	360
" 7	1½	1200	965	774	500
" 8	1½	1560	1280	910	639
" 9	2	1980	1620	1380	810
" 10	2	2450	2000	1580	990
" 12	2½	2870	2880	2275	1440



The Keystone Injector.

The above cut represents the Keystone Injector, class A, which is used for feeding boilers where the water-supply is received from street-mains, reservoirs, cisterns, etc. It should be

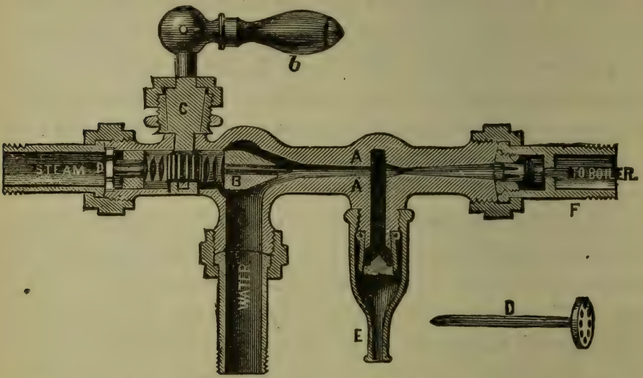
placed in a horizontal position, and, if the water is taken from a tank, the injector should be below the supply. All connections, whether for steam or water, should be of the same internal bore as the nipples on the injector. The steam should be taken from the highest part of the boiler, in order that it may be dry, and the pipes should be as short and straight as possible, and should not be connected with any supply-pipe or feeder employed for any other purpose. A globe-valve should be placed on both the steam- and water-pipes, but no extra check-valve is necessary, except the one in the swivel, which controls the outlet to the boiler; nor is any washer or packing necessary for any part of the injector or its connections, as all of its joints are ground.

To start the injector.—Open the steam-valve for the purpose of allowing any water resulting from the condensation of steam to escape; then close it; next open the water-cock, then the steam-valve, and move the plug *B* slowly forward by means of the handle *b*, until the water ceases to appear at the overflow. And if, while the injector is working, water should commence to run from the overflow, move the plug slowly forward until the water ceases to flow. If steam escapes, move the plug backward for the purpose of giving the injector more water. When the lever *b* is set, so that the injector works dry, all that is necessary to do to stop its feeding is to close the steam-valve first, then the water-valve; and, when it becomes necessary to feed again, the injector may be started by first opening the water-cock and then the steam-valve. The lever being only used to regulate the volume of steam- and water-supply, if the lever moves too loosely, it may be tightened by screwing down the nut on the spindle *C*; if too tight, the nut can be slacked up. This injector will work under ordinary circumstances, but there are other injectors in the market which are immensely superior to them.

The Keystone Lifting Injector.

The cut on page 424 represents the Keystone Lifting Injector, class B. The same instructions for setting up and manipulating class A, Fig. 1, apply to this also, with this exception, that no stop-

cock or valve is necessary on the water-supply. The jet *D* serves to create a vacuum, and assists in carrying the water forward against the boiler-pressure; and, as it is stationary, it is always in a proper position to produce a vacuum, the required volume of steam necessary to force the water into the boiler being obtained



by moving the plug *B*, as in the case of class A. To lift water from a well, open the steam-valve for the purpose of removing the water of condensation; then close it; after which move the plug *B* back against the disc of the jet *D*; then open the steam-valve, and, when the water appears at the overflow, move the plug slowly forward until the water ceases flowing, after which the injector will sometimes lift water, but are said not to be reliable as lifting injectors.

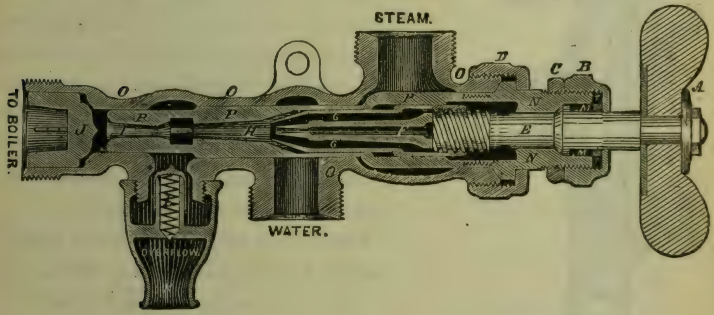
The Eclipse Injector.

The cut on page 425 represents the Eclipse Injector, with Sellers' Lifting Jet, which is claimed to embody many desirable features, such as simplicity, durability, and easy adjustment, and to work under a steam-pressure ranging from 5 to 150 lbs. per square inch, without breaking the water-supply; that, when it becomes necessary to fill a boiler with cold water, all the working parts may be removed from the barrel, which will permit the water to flow through without obstruction; that it will heat the feed-water up

to 200° Fah.; and that it is particularly adapted to heating the water in the tenders of locomotives, to prevent them from freezing.

The same precautions that are necessary to be observed in connecting all injectors, viz., that the steam be taken from the highest point of the boiler; that a valve must be placed in the steam-pipe near the swivel, and also one on the feed-pipe between the boiler and check-valve; and that the water connections are perfectly tight, are applicable to this one, also.

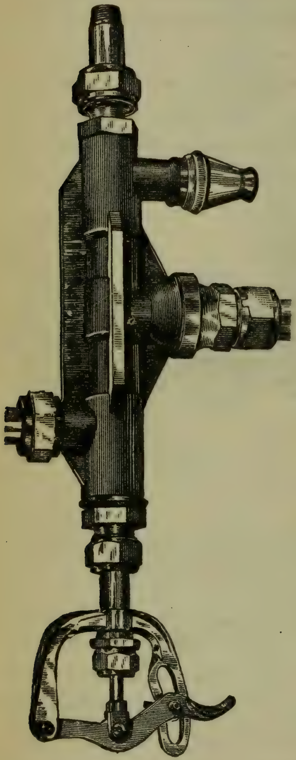
Directions for using the Eclipse Injector.— Close the regulator, *A*, by turning it to the right as far as it will move; then turn on the steam, slowly at first, until the water which is taken up shows at the overflow; next open the regulator slowly, until the dis-



charge from the overflow ceases; the injector will then be working. When it becomes necessary to stop working, first turn off the steam; then close the regulator, as otherwise, when started again, it will not lift quickly; but, when the water flows to the injector from either a hydrant or tank, after the injector has once been adjusted, it is only necessary to turn on the water, and then the steam. To remove the working parts from the barrel of the injector, screw the jam-nut, *C*, up against the main nut, *D*; then, by keeping the jam-nut tight against *D*, the injector may be easily drawn out from the shell. Should it become necessary to repack the injector at *M*, care must be taken to insert the packing in front of the follower, *T*, and compress it with the latter.

The Clipper Injector.

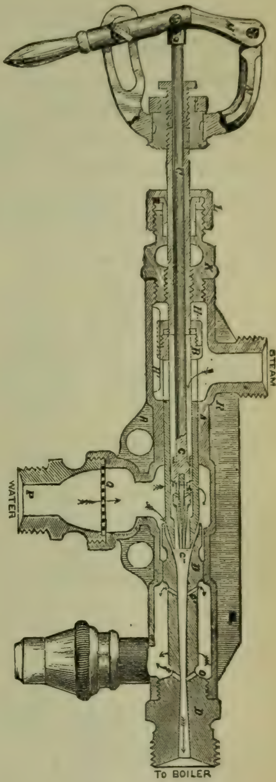
The annexed cut represents the Clipper Adjustable Injector, which is claimed to possess the following good qualities: simplicity of construction, certainty of action, ease of starting, non-liability to get out of order, large capacity, and that it will draw water as far as a siphon or pump, and force it into the boiler under ordinary pressure. Besides, it can be regulated so as to feed one-half its capacity, and will not slip.



All that is necessary to insure certainty of action in this injector, is to place it in a horizontal position, and take the steam from the highest point in the boiler, and to have the steam- and water-pipes fully as large as the openings in the swivels to which they are attached.

The cut on page 427 shows a section of Lynde Clipper Injector.—*A* is the shell or body; *B*, the steam-tube; *C*, the jet or lifting-tube; *D*, the water-tube; *H*, the swivel which is kept from turning by the fins *H*; *K*, the bonnet, by unscrewing which the tubes *B* and *C* may be removed; *M* and *N*, revolving lever and handle by which to regulate the water and steam; *O*, overflow holes; *O′*, holes to assist in lifting, on starting the injector; *Q*, strainer to prevent any foreign substances from entering the injector with the water; *R*, ribs to prevent the shell from springing; *W*, overflow valve and spring.

How to start the injector when the water flows to it.—Draw the steam-tube, *B*, nearly all the way back, by revolving the handle, *M*, which actuates tube, *B* (same as the wheel does the valve in a common globe-valve), and pull lever, *M'*, all the way back. Open steam-valve a little, to clear pipe of condensed water; when steam blows out at overflow, push lever, *M'*, full forward, open steam full, and open water-cock. When water runs solid from overflow, draw lever, *M'*, slowly all the way back, and turn in tube, *B*, slowly till water ceases. The injector is then set to feed its maximum amount at the pressure of steam then used. It may then be started by simply opening steam-valve a little, as above, to clear the pipes; then close steam- and open water-cocks. When water runs solid at overflow, open steam-valve slowly, and feeding will commence without operating lever, *M'*.



How to start the injector when the water is to be lifted.—Draw steam-tube, *B*, nearly all the way back, and pull lever, *M'*, all the way back; open steam-valve a little (or all the way, if desired), to clear steam-pipe, and, when steam appears at overflow, push lever, *M'*, full forward — the water-pipe being open, water will be likely to appear at once (or in a few seconds) at overflow; if not, pull lever, *M'*, back a *moment* to clear the injector and push full forward again. As soon as the water runs solid at overflow,

pull lever, M', slowly all the way back, and screw in tube, B, until feeding commences. It is then feeding the maximum amount at pressure. It may then be started by turning on steam; push lever, M', full forward, and then pull back as above, when water appears at overflow.

To reduce the feed in either case. — When the injector is set as above, push lever, M', forward, until water begins to run from overflow; then cut off water with handle, M, until it ceases at overflow, and repeat as long as it will bear, and continue to feed. The minimum feed is thus obtained, and the water is delivered to the boiler the hottest.

TABLE

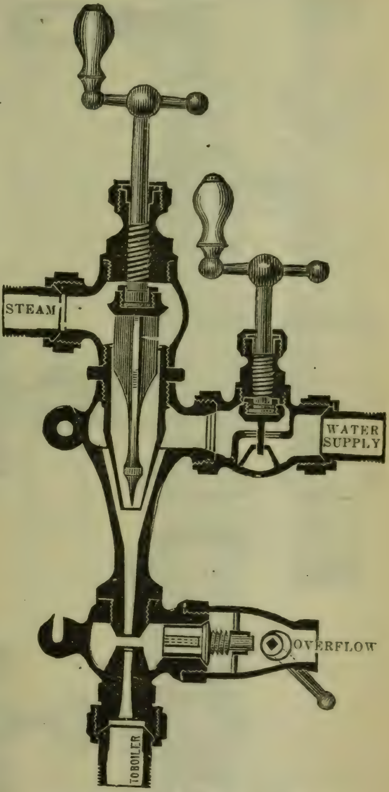
OF CAPACITIES OF CLIPPER INJECTORS.

No.	Pipes.		Approximate Gallons of Water thrown per Hour, with 60 to 100 Lbs. of Steam.
	STEAM.	WATER.	
1	3/8 in.	3/8 in.	60 to 90
2	3/8 "	1/2 "	150 to 180
3	1/2 "	3/4 "	250 to 300
4	3/4 "	1 "	500 to 600
5	1 "	1 1/4 "	700 to 900
6	1 "	1 1/4 "	800 to 1200
7	1 1/4 "	1 1/2 "	1200 to 1600
8	1 1/4 "	1 1/2 "	1600 to 2000
9	1 1/2 "	2 "	2000 to 2500
10	1 1/2 "	2 "	2500 to 3000
12	2 "	2 1/2 "	3000 to 3500

There is one principle that governs the action of all injectors, which is, that if the temperature of the water is raised too high, they will not work. Some injectors will lift water as high as 20 feet, according to the temperature of the water and the size of the injector; large injectors having invariably the greatest lifting capacity. As the amount of water thrown depends on the velocity

of the steam, it follows that the volume of water thrown will be much greater with high than with low steam-pressure.

The annexed cut represents Mack's Fixed-Nozzle Injector, which is said to have a working range, with one handle, of from 15 lbs. to 175 lbs. steam-pressure per square inch, and is always reliable, whether worked constantly or once in a year. When extraordinarily high pressure is required, an extra valve is attached, which will admit of working this injector at a range of 5 lbs. to 250 lbs. per square inch.

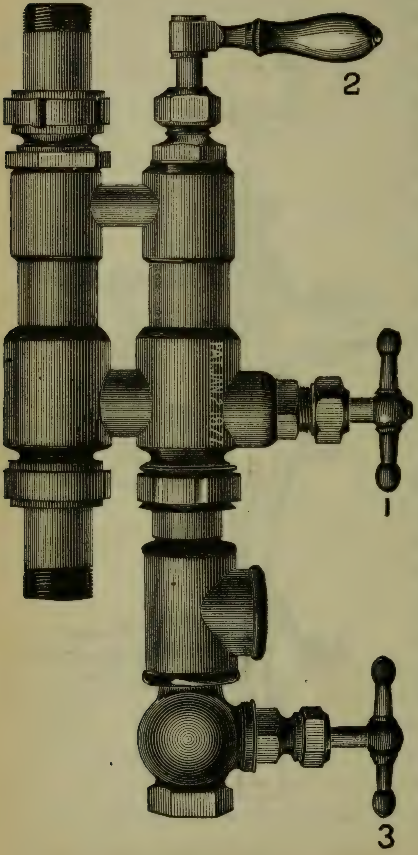


Mack's Fixed-Nozzle Injector.

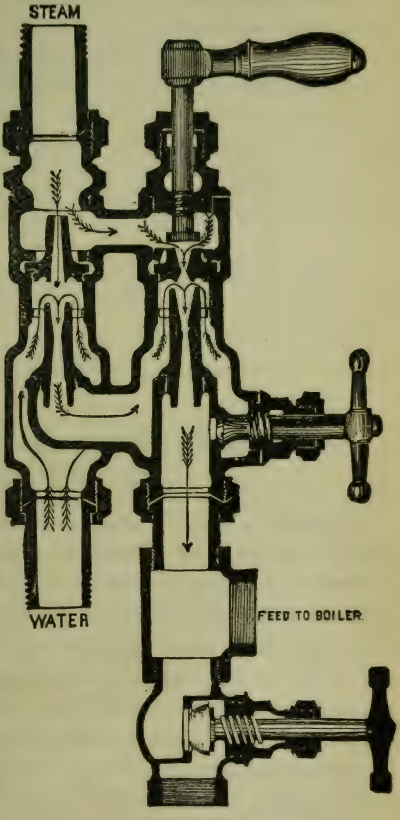
Fixed-Nozzle Injectors have no movable or adjustable parts within them; they can be regulated by steam- and water-supply cocks on the outside of the instruments; but there is one pressure of steam to which they have been primarily adapted, and at which they work best, viz., at the pressure at which they give the largest duty. Inasmuch as the pressure of steam in stationary boilers is, as a rule, held constant, they are well suited for that kind of work; but in cases where there is a great variation of pressure they are not so well suited. There are fewer of them in use than of other arrangements, nevertheless some of them give satisfaction, but in any case their simplicity is their chief recommendation.

The Inspirator.

The inspirator, though belonging to the injector family, differs from the latter, inasmuch as it is a double instrument, consisting of a lifting and a forcing side; the latter being to all intents and purposes, with slight mechanical variations, an "injector," while the former is a kind of a pump, which supplies the forcer side with water. The whole machine is a curious combination of mechanical arrangements for lifting and forcing water, and cannot be actually said to be either an injector or a pump, though it performs the functions of both. The inspirator is capable of lifting and forcing water or other fluids to a great height. It will lift water 25 feet, with a steam-pressure of 30 lbs., provided the suction-pipe be perfectly tight, and the instrument is furnished with dry steam; but the temperature of the water will control to a certain extent the height of the lift. For a lift of 25 feet, the temperature of the water should not exceed 100° Fah.



Whenever the inspirator fails to act, the trouble, in a majority of cases, will be due to leakage in the pipes. Other causes are due to the area of the suction-pipe being too small, which ought in all cases to be larger than the nipple or swivel to which it is connected; but in any case it is advisable to have a foot-or check-valve in the suction-pipe, below the level of the water in the well, river, or mine.



How to operate the inspirator. — When steam is admitted to the inspirator, it passes through the *lifter steam-jet*, leaps the interval, *A*, through the *combining-tube*, and escapes at the overflow, thus expelling the air and producing a partial vacuum, into which the water rises. As soon as the water appears at the overflow, close valve No. 1, to prevent it from escaping, and induce it to pass up the *forcer*, and through the *combining-tube B*; then by opening the handle No. 2, and closing No. 3, the water is forced directly through the feed- or delivery-pipe into the boiler or tank, as the case may be. The inspirator is adapted as a boiler-feeder for either stationary, locomotive, or marine engines.

TABLE

OF CAPACITIES OF THE HANCOCK INSPIRATOR.

Number of Inspirator.	Size of Pipe Connections.	Gallons per Hour.
10	$\frac{1}{2}$	120
15	$\frac{3}{4}$	320
20	1	540
25	$1\frac{1}{4}$	900
30	$1\frac{1}{2}$	1260
35	$1\frac{1}{2}$	1540
40	2	2240
45	2	2820
50	$2\frac{1}{2}$	3480

Instructions for Setting up, Properly Attaching, and Adjusting Injectors.

All pipes, whether steam, water-supply, or delivery, should be of the same internal diameter as the hole in the corresponding branch of the injector, and as short and straight as practicable.

When floating particles of wood, or other matter, are liable to be in the supply-pipe, a strainer should be placed over the receiving end of it. The holes in this strainer should be as small as the smallest opening in the delivery-tube, and the total area of all the holes should be greater than the area of the water-supply pipe, to compensate for the closing of some of them by deposits.

The steam should be taken from the highest part of the boiler, in order to avoid the carrying over of water with the steam; but it should not be taken from the pipe leading to the engine, unless such pipe is large.

When any injector capable of raising water is set, care must be taken to have the pipes as tight as possible, so as not to draw air.

If the water is not lifted by the injector, but flows to it from a tank or hydrant, there should be a cock in the water-supply pipe; and in case the injector be self-adjusting, this cock should be of a kind that will prevent any considerable pressure in the water-supply pipe between it and the injector. The higher the steam is carried in the boiler, the greater may be the pressure in the water-supply pipe.

There should always be a stop-valve or cock in the steam-pipe, between the steam-space in the boiler and the injector, and a check-valve between the water-space of the boiler and the injector.

When an air-chamber is placed below the injector in the water-supply pipe, care should be taken to keep it supplied with air. When the injector lifts water from a tank placed below it, no precaution is needed, as, when the injector is stopped, the water flows back and air enters the pipe.

When fed from a hydrant through a self-regulating valve, there should be a pet-cock between the valve and the air-chamber, which will serve to drain away the water when the valve is closed and the injector is not working.

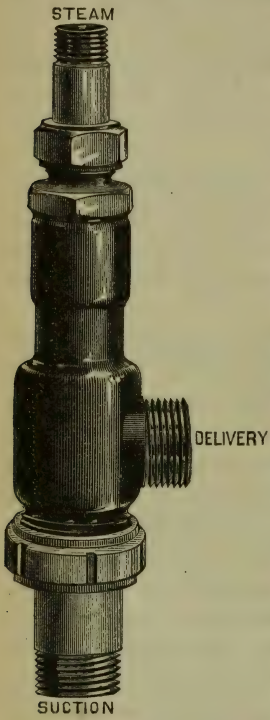
After all the pipes are properly connected to the injector and to the boiler, and it is ready for work, they should be disconnected and well washed out, in order to remove any obstructions, such as paint, red lead, straw, or shavings, that may have found their way into them. Many excellent instruments have been condemned because those who set them up failed to take this precaution.

Injectors, like pumps and other hydraulic machines, are not so reliable in action when working water of a high temperature as when the temperature is moderate; though there are several injectors, owing to peculiarities in their mechanical arrangements, more reliable in this respect than others.

Injectors, like nearly all other machines connected with steam-boilers, are frequently neglected, and allowed to become covered with filth, which, in view of their wonderful utility and efficiency, is a reproach to those who have them in charge.

The Ejector or Lifter.

The annexed cut represents the ejector or lifter, which is practically the lifter side of the inspirator, with a reduced steam-jet and enlarged lifter combining-tube. It is suitable for breweries, tanneries, bleacheries, etc.; for transferring large volumes of water, lye, acid, and other liquids. It will deliver more fluid of any kind at a low lift, with a lower pressure of steam, than either the injector or inspirator; but it is not as reliable, or as well adapted to the different purposes for which these instruments are used, as either of them. It answers a very good purpose when cellars become flooded in consequence of heavy rain-falls, high tides, or overflowing of culverts, and requires no very intelligent management. Its action is based on the same principle as that of the injector, and is more simple, as it has no adjustable or movable parts.

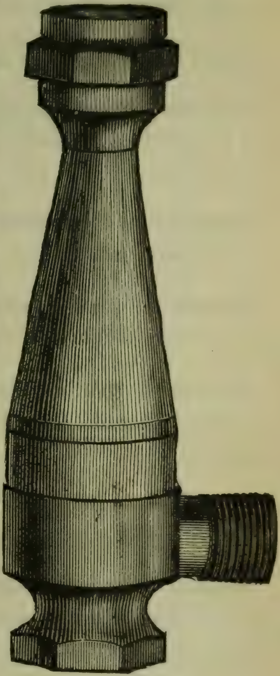


Method of starting the ejector.—All that is necessary to start the ejector is to turn on the steam, after which it will work as long as the water-supply and steam-pressure continue; and it is immaterial what lift it is started on, as the steam-supply may be gradually reduced to meet the requirements of the quantity of water to be discharged. If started on a steam-pressure of 40 lbs. per square inch, it will continue to work until the pressure falls to 15 pounds.

The Ejector and Inspirator are manufactured by the Hancock Inspirator Co., Boston, Mass.

Jamison's Steam Water-Ejector.

The annexed cut represents Jamison's steam water-ejector, as it is termed, which, like other ejectors, is suitable in tanneries, breweries, or places where large quantities of liquids which contain floating particles, such as malt, hops, bark, sawdust, etc., require to be lifted, as it has no moving mechanism to be obstructed or clogged. Its action is based on the same principle as the steam siphon, and when once started, by simply turning on the steam, it will continue to work as long as the steam and water-supply lasts, through a diminution of pressure ranging from 15 to 100 lbs. per square inch, and *vice versâ*. As they are generally made of brass or some non-corrosive metal, they rarely ever wear out or require any attention. They are just as efficient when submerged in water or other liquid as when directly on the surface.



TABLE

OF CAPACITIES OF JAMISON'S STEAM WATER-EJECTOR.

SIZE.	CAPACITY.
⅜ inch ejector.	4 gals. per minute.
½ " "	8 " " "
¾ " "	12 " " "
1 " "	15 " " "
1¼ " "	20 " " "
1½ " "	30 " " "
2 " "	80 " " "
3 " "	300 " " "

Questions,

THE ANSWERS TO WHICH WILL BE FOUND IN THE TEXT.

What is the object of attaching a condenser to a steam-engine?

Give the names, and the advantages and disadvantages of the two kinds of condensers in most general use, with a description of the same.

Explain how the injection-water enters and escapes from surface and jet condensers.

State what relative proportion the jet condenser should bear to the steam-cylinder of a condensing engine.

State what relative proportion the cooling surface in a surface condenser should bear to the cubic contents of the steam-cylinder.

State the respective advantages and disadvantages of having condensers too large or too small.

What is the most advantageous temperature at which to keep the water in hot wells? and what effect does too high or too low a temperature exert on the economical working of the engine?

Explain the arrangements by which the bilge injection-water is introduced into jet and surface condensers.

What would be the effect of not shutting off the injection-water when the engine is stopped?

State the quantity of water necessary to condense steam, with a formula.

Give the rule for finding the cooling surface in the tubes of surface condensers.

What is the most practicable method of cleaning the tubes of surface condensers when they become foul?

State what relative proportions the circulating-pump should bear to the steam-cylinder of a surface-condensing engine.

Explain the principles involved in the working of the Korting jet condenser; also the method of starting it.

What is the meaning of the term vacuum?

How is the vacuum maintained in the condenser of a condensing engine?

What effect has the temperature of the injection-water on the vacuum?

How is the vacuum measured?

Suppose the steam-gauge shows 60 lbs. pressure, and the vacuum-gauge registers 26 inches, what will be the effective pressure on the piston?

Why does the condensation of steam produce a vacuum in the condenser?

How is the state of the vacuum shown?

From what causes is an imperfect vacuum most likely to arise?

How would you proceed to discover the cause of an imperfect vacuum?

Is a vacuum power?

Can a perfect vacuum be maintained? If not, why not?

What is the object of air-pumps used in connection with condensing engines?

What relative proportion should the air-pump bear to the steam-cylinders of simple and compound surface-condensing engines?

What relative proportions should the air-pump bear to the steam-cylinders of jet-condensing engines?

What is the difference in the duty which the air-pumps of surface-condensing and jet-condensing engines have to perform?

Explain the difference between bucket, piston, plunger, single and double acting air-pumps.

What is the object of attaching an air-valve to a circulating, reciprocating, or double acting pump?

What is the most probable cause of an air-pump with a foul valve, and no discharge-valve, failing to work?

What is the object of an air-pump trunk?

What are the functions of an air-pump pet-cock?

Describe the construction of an air-pump bucket.

With what metals are air-pump rods generally covered, and why are they so covered?

Give the shape and functions of a ship's side air-pump discharge-valve.

What is the object of an air-casing?

What is the object of the mariner's compass?

What are the causes of variation of the compass?

What is the meaning of the term rhumbs?

What is the equator?

What are the poles?

What is a meridian?

What is the meaning of the term latitude?

What is understood by difference of latitude?

What is the meaning of the term departure in its relation to navigation?

What is the meaning of the term longitude?

What are degrees of longitude?

Give the rule for reducing degrees of longitude to time.

What is the difference of longitude between any two places?

Define the term distance in its relation to navigation.

Define the terms course and magnetic course as applied to navigation; also the terms true course and course made good.

Define the terms variation, deviation, and error of the compass.

Define the term leeway.

Define the terms meridian; and also apparent, observed, and true altitude.

Explain the term visible horizon and dip of the horizon.

What is the meaning of the term refraction?

Give the meaning of the term port side.

What is the parallax?

What is the meaning of the term declination?

What is meant by polar distance?

What is meant by right ascension?

What is meant by semi-diameter?

Give the meaning of the term starboard side.

What is meant by the augmentation of the moon's semi-diameter?

What is the zenith distance?

Explain the terms civil, astronomical, sidereal, apparent, and mean time; also the equation of time.

What is meant by the hour-angle of a celestial object?

Define the terms ecliptic and the tropics.

Define the term azimuth.

What is meant by the term amplitude when applied to navigation and astronomy?

What is the meaning of the term dead reckoning?

Give the sailing distance from New York, in geographical miles, to different ports.

Give the latitude and longitude of different seaports.

To what class of machines do pumps belong?

What principle is involved in the working of all pumps?

How high will an ordinary pump, in good condition, lift water or other liquids?

What condition limits the action of all atmospheric pumps?

Give the rule for finding the size of pump-plunger and stroke for an engine of any given power.

Give the rule for finding the quantity of water, or other liquid, that any pump will lift or discharge in a given time.

Give the most probable causes why pumps fail to work satisfactorily.

Explain the difference between lift-force, single-acting, and double-acting pumps.

What is the meaning of the term circulating-pump?

What is the object of placing an air-chamber on a pump?

Give the rule for finding the power required to raise a given quantity of water.

Give the reason why pumps will not lift very hot water.

What is the object of placing a pet-cock on the barrel of a feed-pump?

What is the object of placing mud-boxes or strainers on the suction-pipes of pumps?

What course would you adopt to prevent pump-pipes from freezing in cold weather?

Explain the meaning of the terms injector and ejector when applied to hydraulic machines.

Explain the principles involved in the working of the injector.

What conditions limit the height to which injectors can lift water?

Under what three heads may all injectors be classed?

What are the meanings of the terms maximum and minimum delivery?

What is the meaning of the term range when applied to injectors?

To what class of machines does the inspirator belong?

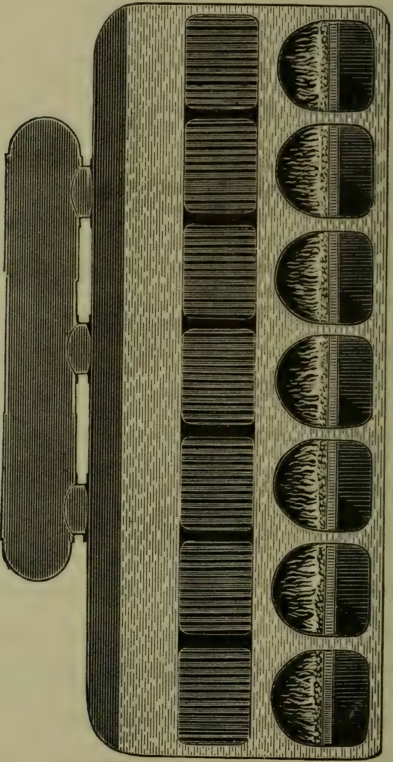
On what principle is the action of the inspirator based?

Explain the advantages and disadvantages of injectors, ejectors, and inspirators over pumps.

PART SIXTH.

Steam-Boilers.

Steam-boilers embrace a great variety of designs;* in fact, any vessel in which steam is generated for mechanical purposes

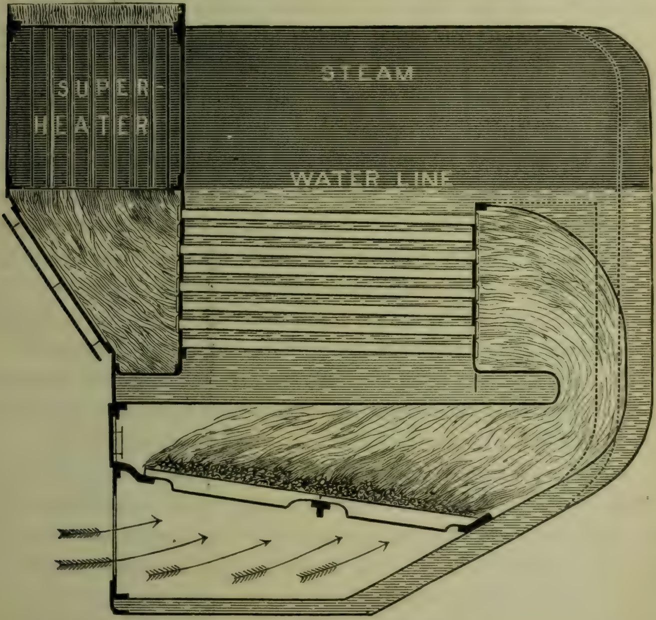


Water-Tubular Marine-Boiler.

may be termed a steam-boiler, regardless of shape or form. The

* For a full description of all the steam-boilers in use at the present day, their peculiarities of design, construction, care, and management, see Roper's "Use and Abuse of the Steam-Boiler."

most common forms of marine-boilers in use at the present day are the horizontal and vertical, fire- and water-tubulars. The water-tubular is fast disappearing, and is now rarely to be found except in the United States Navy, or those of other countries. Its gradual disappearance arises from the fact that it is more expensive to build and to repair, is more dangerous, and requires extra care and management. If a tube splits or becomes leaky in the fire-tubular boiler, the difficulty may be met by plugging, and the vessel can proceed on its way; but if the same accident occur in a water-tubular, it would be necessary to blow out the boiler. The same principle which was embodied in the Montgomery water-tubular marine-boiler was



Fire-Tubular Marine-Boiler.

introduced into the Dimpfel locomotive-boiler, but soon fell into disuse in both cases. The fire-box, fire-tubular marine-boiler, with

combustion-chamber at the back end and superheater in the uptake, is the type of boiler most generally in use on the steamships of the different lines sailing out from the seaports of this country as well as those of other nations.

Aside from the choice among engineers between the two forms, there is a wider difference in their proportion than in anything else connected with the steam-engine. While all generally agree that, in proportioning a marine-boiler, there should be sufficient grate-surface to consume the maximum quantity of coal required for the engine for which that boiler was intended to furnish steam, and that there should be sufficient heating-surface to absorb the heat evolved by the fuel; yet, when it comes to laying down proportions, one engineer allows twice as many square feet of heating-surface to one square foot of grate-surface as another. Watt's proportions for land- and marine-boilers varied from 9·5 to 10 feet of heating-surface to 1 square foot of grate-surface. Maudsley and Miller allowed 10 square feet of heating-surface to 1 square foot of grate-surface in the boilers of the celebrated ocean steamer Great Western, and from 10 to 12 square feet of heating-surface to 1 square foot of grate-surface in other marine-boilers that they constructed about the same time; so that neither they nor Watt seemed to have any fixed rule, nor did there appear to be any among naval constructors either in this country or England.

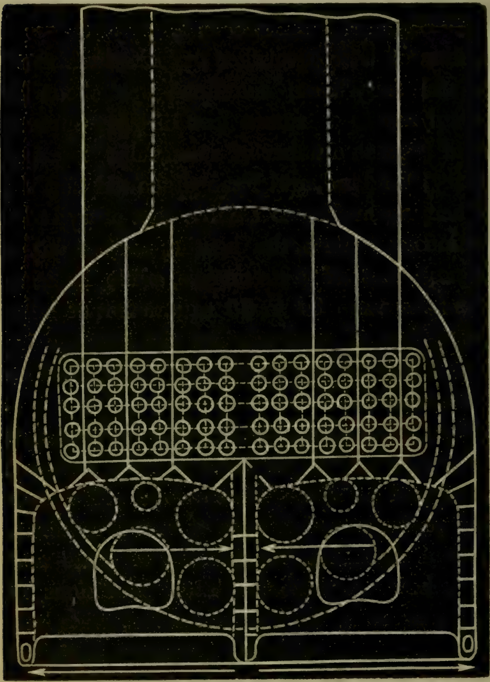
This may be seen from the fact that the U. S. gun-boat Massachusetts had 34 feet of heating-surface to 1 square foot of grate-surface, while the Vixen, with the same-sized engine, had only 16 to 1. The merchant-steamer Constitution had 66 square feet of heating-surface to one square foot of grate-surface, while the Franklin, a steamship of nearly the same capacity, with engines of the same power, had only 28 to 1. The boilers of the celebrated steamships of the Collins Line, which have made such fast time between New York and Liverpool, had 33 square feet of heating-surface to 1 square foot of grate-surface, while in the boilers of the steamships of the Cunard Line the heating-surface varies from 18 to 37 square feet to 1 square foot of grate-surface. The Mary Powell,

one of the fastest river-boats in American waters, has 17 square feet of heating-surface to 1 square foot of grate-surface. In proportioning the heating-surface to the cubic contents of the cylinder, the same variation seems to exist which shows there is no recognized proportion for either. The steamship Massachusetts, U. S. N., has 77 square feet of heating-surface to 1 cubic foot of cylinder, while the Powhatan has less than 15 square feet, and the San Jacinto has a trifle over 12. The merchant-steamer Union had one hundred and eighteen square feet of heating-surface to 1 cubic foot of cylinder, while the Isaac Newton had only 10 to 1. The steam-tug Rescue had 63 square feet of heating-surface to 1 cubic foot of cylinder, while the Anglo-Saxon had only 10 to 1.

The average proportion of heating-surface to grate-surface of 345 steamships, tugs, and ferry-boats examined was about 30 square feet of heating-surface to 1 square foot of grate-surface, while an examination of a great number of steamships, tug, and ferry-boats in this country, England, and France, showed that the average proportion of heating-surface to 1 cubic foot of cylinder was about 28. In stationary boilers the heating-surface varies from 12 to 30 to 1 square foot of grate-surface, while in some patented sectional boilers there are 60 to 70 square feet of heating-surface to one square foot of grate-surface, the average for locomotive-boilers being about 60 square feet of heating-surface to 1 square foot of grate-surface.

To proportion a marine-boiler understandingly, it is necessary to know the size of the engine and of the boat or ship, the load to be propelled, and the speed at which it is to move. The engineer can determine the pressure and volume of steam required, and decide on the degree of expansion, the quantity of grate- and heating-surface, and in relation to these two latter conditions, as shown in the foregoing paragraphs, the field has a very wide latitude. But he must be sure that the boiler possesses sufficient strength to resist in safety the maximum pressure to which it will ever be exposed; that it contains sufficient grate-surface for the combustion of the necessary quantity of fuel under any circum-

stances; that it has sufficient heating-surface to evaporate the necessary quantity of water; that it is capable of containing a sufficient supply of water and steam to prevent undue fluctuation, and that it affords convenient facilities for the repair or renewal of any of its parts. After the foregoing conditions are determined on, another object of great importance to be considered is making the boiler as light and compact as possible. The term heating-surface, when applied to steam-boilers, means all that part of the fire-box, crown-sheet, tube-sheets, and flues with which the fire and flame come in contact in their escape from the furnace to the chimney.


Direct Flue and Return Tubular Marine-Boiler.

Martin's upright tubular-boiler is sometimes used for marine purposes. Its only advantage is economy of space; its first cost is more than that of the ordinary horizontal marine tubular-boiler, and it is not more efficient. The capacity of the steam-room is about one-third the capacity of the boiler.

The quantity of steam that can be generated in any boiler in a given time is dependent upon a great variety of circumstances, such as the kind of boiler, its condition as to dirt, scale, etc., the

manner in which it is set and fired, the quality of the fuel used, quantity of grate-surface, amount of heating-surface, draught, etc., while the amount of water used will depend entirely on the engine, provided the steam is dry. The evaporation in tubular boilers,—stationary, locomotive, and marine,—under good conditions, is about $8\frac{1}{2}$ to 9 lbs. of water to 1 lb. of coal; in flue-boilers, 6 to 7; but the average result is about 25 per cent. below this. The nominal loss of fuel in boilers is rarely less than 30 per cent., and is frequently as high as 50. Taking the lowest estimate at 30 per cent., it may be illustrated as follows: The amount necessary to produce a draught, including the flame which escapes into the chimney, 20 per cent.; particles of coal falling through the grates, 5 per cent.; loss arising from the formation of carbonic oxide, 3 per cent.; loss induced by radiation, 2 per cent.

The common estimate of the quantity of water necessary to produce one horse-power is one cubic foot; the amount of heating-surface necessary to evaporate one cubic foot of water in an hour has been found, by experiment, to be about 14 square feet to $\frac{1}{2}$ square foot of grate-surface, under the most favorable conditions. It has grown into a custom, in estimating the horse-power of steam-boilers, to allow 14 square feet of heating-surface to $\frac{1}{2}$ square foot of grate-surface; but the evaporative performance of steam-boilers varies very much, as in one boiler a cubic foot of water may be evaporated in an hour by 9 square feet of heating-surface to $\frac{1}{2}$ square foot of grate-surface, while another will take double the amount. In locomotives, the proportion of heating-surface to grate-surface is about 50 to 1; in marine-boilers, about 28 to 1; ordinary cylinder-boilers, about 15 to 1; flue-boilers, 18 to 1; tubular-boilers, from 20 to 24 to 1; and in sectional- or patent-boilers, about 30 to 1.

The tendency of water to foam in marine-boilers is frequently attributed to the presence of dirt, or other saline matter, in the water; but it is often induced by want of proper relations between the heating-surface, steam-room, and water-space of the boiler, as, when the discharge of steam is out of proportion to the steam-room

in the boiler, the high temperature required to generate steam with sufficient rapidity to supply the demand causes violent boiling, and the agitation is greater when the relation between the temperature and pressure is most disturbed. This is often the case with tug-boats just starting to tow a heavy vessel. Boilers with a large amount of heating-surface and small steam-room generally foam.

Marine-boilers are generally surmounted by a dome, and, though domes do not add much to the cubical capacity of the steam-room, they have the effect of superheating the steam, or imparting to it an extra heat, which greatly increases its expansive force, and renders it less liable to condense in the passages between the boiler and the cylinder.

Fittings of marine-boilers.—The fittings of marine-boilers are the funnels, air-casings, uptakes, smoke-box and fire-doors, grate-bars, bearers and bridges, main steam-pipe and stop-valve, donkey-valve, safety-valves and drain-pipes, main- and donkey-feed check-valves, blow-off- and scum-cocks, water-gauges, test water-cock, steam-valves for whistle, and winches.

Bursting Pressure of Cylindrical Steam-Boilers.

The force which will rupture a cylindrical boiler depends upon the diameter and the pressure of the steam ; hence, the total pressure to be sustained is equal to the diameter, multiplied by the pressure per square inch of surface, multiplied by the length of the boiler. The shorter the tube, and the smaller the diameter, the greater its power of resistance, and *vice versâ.* No matter what the diameter of a boiler may be, the transverse, or cross pressure tending to tear it asunder, will always be double the longitudinal pressure.

Rule *for finding the bursting pressure of cylindrical boilers with riveted seams.*

Multiply the tensile strength of the iron (which should be taken at 50,000 lbs. per square inch of section) by ·56, if single riveted,

and by ·70, if double riveted, and divide by the diameter of the boiler, multiplied by the number of pieces of metal, that would make one square inch of cross section; the product will be the bursting strain.

For instance, what pressure will it require to rupture a cylindrical boiler with riveted seams, diameter 12 inches, thickness of iron $\frac{1}{4}$ inch?

$\frac{50,000 \times \cdot 56}{12 \times 4} = 583\cdot 33 \times 2 = 1166\cdot 66$ lbs., about one-fifth of which would be the safe working-pressure.

Rule *for finding the strain exerted in a longitudinal direction by the pressure of steam in a boiler.*

Multiply the area of the head by the pressure in pounds per square inch, and *divide* the product by the circumference of the boiler, and by the number of thicknesses of iron that would make one square inch of cross section; the quotient will be the strain.

Example. Diameter, 12 inches. Area, 113·09 square inches. Pressure, $1166\frac{2}{3}$ lbs.

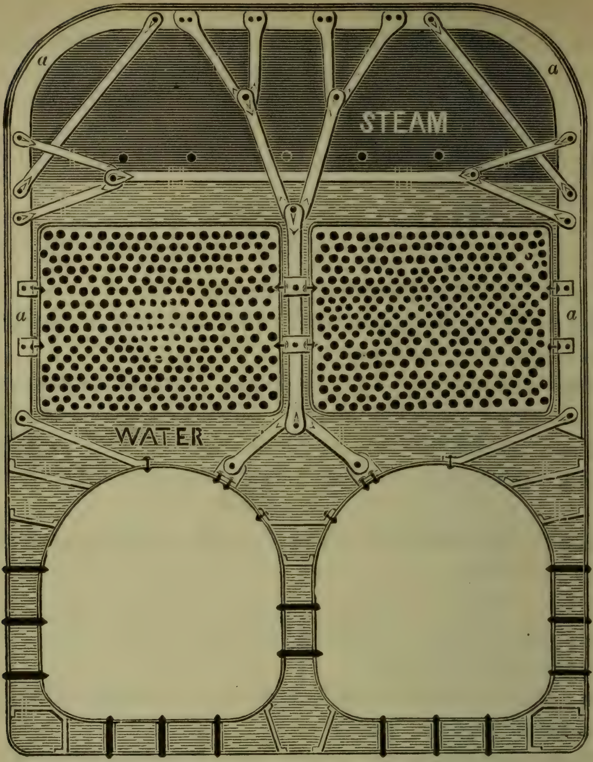
$\frac{113\cdot 09 \times 1166\cdot 66}{37\cdot 69 \div 4} = 14014$ lbs. per square inch of sectional area in a longitudinal direction.

Rule *for finding the strain exerted in a transverse direction by the pressure of steam in a boiler.*

Multiply the pressure per square inch by the diameter, and also by the number of thicknesses of metal it will take to make one square inch of cross section, and *divide* the product by 2, because the boiler has 2 sides.

Example. $\frac{1166\cdot 66 \times 12 \times 4}{2} = 31999\cdot 84$ lbs. per square inch of sectional area in a transverse direction.

The power of any steam-boiler to resist strain depends upon the thickness and quality of material, character of the workmanship, and the shape of the parts subjected to strain.



The above cut represents the arrangements most generally employed for bracing marine steam-boilers, and includes the vertical and horizontal, angle, toggle, dome, and crown braces; as well as the buckles, crow-feet, angle-irons, girths, stay-bolts, and leg braces. The tubes answer for braces for the tube-sheets; the crow-feet for the crown and dome; the proper strength for the braces of marine-boilers may be found by multiplying the number of square inches exposed to the pressure of the steam by six times the steam-pressure to be carried.

Rules.

Rule *for finding the safe working-pressure of iron boilers.* — Multiply the thickness of iron by ·56,* if single riveted, and ·70, if double riveted; multiply this product by 10,000 (safe load): then divide this last product by the external radius (less thickness of iron); the quotient will be the safe working-pressure in pounds per square inch, which, if multiplied by 5, would give the bursting pressure.

In the foregoing rule, the tensile strength of the iron is taken at 50,000, as it has been repeatedly proved by experiment that boiler-plate possesses less tenacity than the same iron would have if rolled into bars.

Rule *for finding the internal strain to which boilers are subjected when under pressure.* — Multiply the surface of the plate required for one square inch of sectional area by the pressure of steam in lbs. per square inch; multiply this result by the diameter of the boiler in inches, and divide by 2, which gives the strain per square inch of sectional area to which the boiler is subjected.

The surface of boiler-plate required for one square inch of sectional area will depend upon the thickness of plate; thus, iron ¼ inch thick will require 4 superficial inches to make one square inch of sectional area; iron ½ inch thick will require 2, and so on.

Rule *for finding the pressure per square inch of sectional area on the crown-sheets of steam-boilers.* — *Multiply* the width of the crown-sheet in inches by its length in inches; *multiply* this product by the pressure of the steam in lbs. per square inch by the gauge; *divide* by 2, if ½ inch iron, and so on according to the thickness.

Rule *for finding the aggregate strain caused by the pressure of steam on the shells of boilers.* — *Multiply* the circumference in inches by the length in inches; *multiply* that product by the pressure in pounds per square inch. The result will be the aggregate pressure on the shell of the boiler.

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* Multiplied by ·56, because the iron loses 44 per cent. of its strength in the process of punching. Double-riveted seams equal ·70 of the original strength.

Rule *for finding the safe external pressure on boiler-flues.* — *Multiply* the square of the thickness of the iron by the constant whole number 806,300; *divide* this product by the diameter of the flues in inches; *divide* the quotient by the length of the flue in feet; *divide* this quotient by 3: the result will be the safe working-pressure.

Rule *for finding the collapsing pressure of boiler-flues.* — *Multiply* the square of the thickness of the iron, in thirty seconds of an inch, by the constant number 262·4; *divide* this product by the length of the flue in feet; *divide* this quotient by the diameter of the flue in quarter feet, and the quotient will be the collapsing pressure in pounds per square inch.

Rule *for finding the number of square feet of heating-surface in a tube, or any number of tubes.*

Multiply the circumference of the tube in inches by its length in inches, and *divide by* 144; the quotient will be the number of square feet of heating-surface. This, *multiplied* by the whole number of tubes, will give the aggregate amount of heating-surface.

Rule *for finding the strength of single- or double-riveted seams.*

Multiply the area of the metal between the holes, in square inches, by the ultimate strength of the metal after punching the holes. The product will be the strength of the seam. Single-riveted seams being equal to about 56 per cent. of the original strength, and double-riveted, 70 per cent.

Rule *for finding the strain due to the pressure of the steam on boiler-stays.*

Multiply the area in inches between the stays by the pressure in pounds per square inch. The product will be the strain in pounds per square inch.

Rules.

Rule *for finding the heating-surface of fire-box boilers — locomotive, marine, or stationary.* — Multiply the length of the furnace-plates in inches by their height above the grate in inches; multiply the width of the ends in inches by their height in inches; multiply the length of the crown-sheet in inches by its width in inches;

also the combined circumference of all the tubes in inches by their length in inches; from the sum of these four products subtract the combined area of all the tubes and the fire-door; divide the remainder by 144, and the quotient will be the number of square feet of heating-surface.

Rule for flue-boilers. — Multiply $\frac{2}{3}$ of the circumference of the shell in inches by its length in inches; multiply the combined circumference of all the flues in inches by their length in inches; divide the sum of these two products by 144, and the quotient will be the number of square feet of heating-surface.

Rule for cylinder-boilers. — Multiply $\frac{2}{3}$ of the circumference of the shell in inches by its length in inches, divide by 144, and the quotient will be the number of square feet of heating-surface.

Rule for tubular-boilers. — Multiply $\frac{2}{3}$ of the circumference of the shell in inches by its length in inches; multiply the combined circumference of all the tubes in inches by their length in inches. To the sum of these two products add $\frac{2}{3}$ the area of both tube-sheets; from this sum subtract the combined area of all the tubes; divide the remainder by 144, and the quotient will be the number of square feet of heating-surface.

Rule *for finding the heating-surface of vertical tubular boilers, such as are generally used for fire-engines.* — Multiply the circumference of the fire-box in inches by its height above the grate in inches. Multiply the combined circumference of all the tubes in inches by their length in inches, and to these two products add the area of the lower tube- or crown-sheet, and from this sum subtract the area of all the tubes, and divide by 144. The quotient will be the number of square feet of heating-surface in the boiler.

Boiler-Stays.

Boiler-stays, in any case, are but substitutes for real strength in the shell or other parts of the boiler. The strain usually allowed on them per square inch is about 5000 lbs. The most common method of securing them is to cut a thread on both ends,

and screw and cold-rivet them into the plates; another method is to flatten the ends of the stay, and secure them to the boiler by means of one or two rivets; still another is to rivet eye-bolts into the shell of the boiler, fork the end of the stay-bolt, and attach to the eye-bolt by means of a cotter. But all the foregoing methods have their objections, as, when the stays become slack, and it becomes necessary to make them taut, the necessity of cutting away the rivets, destroying the thread, and weakening the boiler is necessarily involved.

The most modern and permanent method of securing stays to the shell or ends of steam-boilers is by riveting angle-irons to the parts to be braced, as shown at *a*, *a*, *a*, *a*, in the cut on page 450, and drilling holes in the angle-irons where the brace is to be attached. Then the rods may be forked, and attached to the angle-irons by means of a cotter, which term means a blank bolt, with a splint in its end, which may be expanded with a cold-chisel, to prevent them from coming out. This arrangement has this advantage, that, where the braces become slack, they may be made taut by taking them out, heating them in a forge, and upsetting them. The value of stays as a substitute for strength and safety depends very materially not only on the manner in which they are attached to the parts they are intended to strengthen, but also on their position, which affects their ability to stand tensile strain and compression-pressure. If the stay is properly anchored, it will stand, on a straight pull, a resistance equal to its tensile strength, or it will resist the force of compression equal to its crushing strength; but if it stands slightly oblique, its power of resistance will be very much diminished.

Stay-Bolts.

Stay-bolts are the means usually employed to strengthen the flat surfaces in the fire-box and water-legs of locomotives and marine-boilers; they are generally screwed into both plates, on each side of the water space, and riveted by the process called

cold riveting. Numerous ordinary-sized bolts are preferable to a few large ones. The difficulty in the case of stay-bolts does not arise ordinarily from tensile strength, brought upon the bolt by the steam-pressure, but from relative changes in position of the two sheets through which the bolt passes, caused by a difference in the temperature of the two sheets, and the consequent difference in expansion. For instance, if the side sheet of a fire-box of a locomotive- or marine-boiler expands in a vertical direction ⅛ of an inch more than the outside sheet, then all bolts in the top row will have their inner ends forced upwards from their original position to that extent, and the boilers must spring or bend accordingly; whereas, when both sheets become again of the same temperature, the ends of the bolts are drawn back to their original position.

TABLE

SHOWING THE BREAKING STRAIN OF IRON AND COPPER STAY-BOLTS.

	Breaking Weight in Pounds.	Strength distributed over 25 inches area would give Lbs. per square inch.	Strength distributed over 16 inches area would give Lbs. per square inch.
1. Iron into iron screwed and riveted	25,000	1,000	1,563
2. Iron into copper screwed and riveted	21,400	856	1,338
3. Iron into copper screwed only	16,200	648	1,013
4. Copper into copper screwed and riveted	14,400	576	900

Scale in Steam-Boilers.

Marine-boilers using sea-water require to be frequently blown out to prevent incrustation, or deposit of salt, on their heating

surfaces, which lie between the iron and the water. It not only causes an increased consumption of coal, but allows the iron to become crystallized and burned. The evil effects of the scale are due to the fact that it is a non-conductor of heat. Its conducting power, compared with that of iron, is as 1 to 35·5. Consequently, more fuel is required to heat water in an incrusted boiler than in the same boiler if clean. A scale $\frac{1}{16}$ inch thick will require the extra expenditure of 15 per cent. more fuel; this ratio increases as the scale thickens. Thus, when it is $\frac{1}{4}$ inch thick, 60 per cent. more fuel is needed; $\frac{1}{2}$ inch thick, 150 per cent., and so on; consequently, to raise water in a boiler to any given heat, the fire-surface of the boiler must be heated to a temperature in accordance with the thickness of the scale.

To raise steam to a pressure of ninety pounds, the water must be heated to about 320° Fah. In a clean boiler of $\frac{1}{4}$ inch iron, this may be done by heating the external surface of the shell to about 325°. If $\frac{1}{2}$ inch of scale intervenes between the shell and the water, such is its non-conducting power, that it will be necessary to heat the fire-surface to about 700°, almost red heat. Now, the higher the temperature at which iron is kept, the more rapidly it oxidizes, and at any heat above 600° it very soon becomes granular and brittle, and is liable to bulge, crack, or otherwise give way to the internal pressure. This condition predisposes the boiler to explosions, and makes necessary expensive repairs. Again, it is readily seen that the presence of scale renders slower and more difficult the raising, maintaining, and lowering of steam.

The principal ingredient in the scale which forms in marine boilers using sea-water is sulphate of lime, but no very injurious effect will take place in boilers if the degree of saltness is not allowed to exceed $\frac{4}{32}$. In fact, a thin coat of scale is beneficial, as it protects the iron from corrosion and internal grooving.

Lord's Boiler Compound appears to be the only chemical preparation in use at the present day that will prevent the formation of scale, or remove it after it has been formed, in any class of boilers, whether stationary, locomotive, or marine, as it neutral-

izes the action of the natural chemical salts which form the basis of all scale and incrustation.

An analysis of sea-water shows the relative quantities of the ingredients it contains.

Water	964·745
Chloride of Sodium	27·059
Chloride of Potassium	0·766
Chloride of Magnesium	3·666
Bromide of Magnesium	0·029
Sulphate of Magnesia	2·296
Sulphate of Lime	1·406
Carbonate of Lime	0·033

The minerals which constitute the basis of the scale which forms in steam-boilers using fresh-water from wells, lakes, or rivers, are sulphate of lime, phosphate of lime, carbonate of lime, magnesia, silica, and alumina, with small quantities of sesquioxide of iron, baryta, carbonic acid, organic matter, chlorine, sulphuric acid, potassa, calcium, soda, phosphoric acid, magnesium, etc. The remedies for the prevention and removal of scale from steam-boilers are almost innumerable.

Foaming in Marine-Boilers.

Foaming in marine-boilers using jet-condensers is generally caused by changing the water from salt to fresh, or *vice versâ*, and is made evident by the boiling up of the water in the glass gauge. When foaming arises from this cause, the water in the boiler should be changed as soon as possible, which can be done by putting on a strong feed, and blowing out continuously, or at short intervals; it may even become necessary to throttle down the engine, cut off short, or even stop, in order to ascertain the level of the water in the boilers.

Violent foaming can be checked by opening the furnace-door, closing the damper, and covering the fire with fresh coal; but this

means of relief should be used as little as possible, because it has a tendency to injure the boiler, owing to the sudden contraction of the parts most exposed to the fire. All the phenomena connected with foaming have not yet been satisfactorily explained; but, from whatever cause it may arise, it is always attended with a certain amount of danger. *Foaming* is sometimes confounded with *priming*, but they arise from different causes, and are productive of different results. *Foaming* is always made manifest by the violent agitation, the rising and falling of the water in the gauge, and the muddy appearance of the water.

Foaming is induced in stationary boilers by a filthy condition, particularly in those to which the feed-water is supplied through open heaters, in consequence of the oil or tallow employed for lubricating the cylinder being carried over with the exhaust-steam. The water in locomotive-boilers foams on some parts of the road, while on other sections this phenomenon never manifests itself, which may be attributed to the presence of alkali or saline matter in the water with which the boilers are supplied on certain parts of the road. Foaming is induced in all boilers by the want of proper proportion between the water-space, heating-surface, and steam-room of the boiler, and also from the absence of sufficient steam-room in the boiler to supply the cylinder.

Priming.

The term Priming is understood by engineers to mean the passage of water from the boiler to the steam-cylinder in the shape of spray instead of vapor. It may go on unseen, but it is generally made manifest by the white appearance of the steam as it issues from the exhaust-pipe; as saturated steam, or steam containing water, has a white appearance, and descends in the shape of mist; while dry steam has a bluish color, and floats away in the atmosphere. Priming also makes itself known by a clicking in the cylinder, which is caused by the piston striking the water against the cylinder-head at each end of the stroke.

Priming is generally induced by a want of sufficient steam-room

in the boiler, the water being carried too high, or the steam-pipe being too small for the cylinder, which would cause the steam in the boiler to rush out so rapidly that, every time the valve opened, it would induce a disturbance, and cause the water to rush over into the cylinder with the steam.

The following table shows the result of a series of experiments, carried out by Captain Rodman, for the purpose of demonstrating the effects of sudden strains on wrought-iron, a bar one inch square, of the best quality of iron, being selected for the purpose.

AMOUNT OF STRAIN.	Temp'y Stretch. $\frac{100}{1000}$ of an inch.	Permanent Stretch. $\frac{100}{1000}$ of an inch.
5,000 lbs.	20	0
10,000 "	41	1
15,000 "	57	1
20,000 "	76	3
25,000 "	100	7
30,000 "	537	408
35,000 "	1833	1661
40,000 "	4000	
45,000 " broke.............		

It will be seen from the above table that the first essay, by means of a strain of 5000 lbs., produced no permanent stretch in the bar; and that 10,000 lbs. and 15,000 lbs., respectively, only produced a permanent stretch of $\frac{100}{1000}$ of an inch, or about $\frac{1}{50}$ of the temporary stretch. But in the next two strains of 20,000 and 25,000 lbs., the iron begins to shows a great acceleration of the weakening process or increase of fatigue, as the permanent strain has sprung up to $\frac{1}{15}$ of the entire stretch. In the next two items this acceleration is astounding, the permanent stretch being $\frac{3}{4}$ of the whole upon 30,000 lbs, and $\frac{9}{10}$ of the permanent stretch of 35,000 lbs. The tensile strength of good boiler-iron increases with an increase of temperature up to about 500° Fah., consequently, a steam-boiler is safer and stronger under a moderately high steam-pressure than it would be under the same hydraulic pressure.

Deterioration of steam-boilers.—Deterioration of steam-boilers arises from the following causes: want of lamination in the sheets; overstretching of the fibre of the plate in the process of rolling; injuries done the plate in the process of punching; damage induced by the use of the drift-pin; injury done the plates by a want of skill in the use of the hammer, or in the processes of hand-riveting and calking. Other causes are unequal expansion and contraction, resulting from a want of skill in setting; grooving in the vicinity of the seams; internal and external corrosion; blowing out the boiler when under a high pressure, and filling it again with cold water when hot; allowing the fire to burn too rapidly after starting, when the boiler is cold; ignorance of the use of the pick in the process of scaling and cleaning; incapacity of the safety-valve; excessive firing; urging or taxing the boiler beyond its safe and easy working capacity; allowing the water to become low, thus causing undue expansion; deposits of scale accumulating on the parts exposed to the direct action of the fire, thereby burning or crystallizing the sheets or shell; and wasting of the material by leakage, etc.

Corrosion, and its Analogy to Combustion.*

The term corrosion means wasting, pitting, or grooving of the material, and is generally referred to under two heads, namely, internal and external.

Internal corrosion presents itself in various forms, and is due to various causes, but principally to the minerals and acids contained in the feed-water with which steam-boilers are supplied.

External corrosion is said to be due to the galvanic action of the mineral in the fuel and the gases in the atmosphere, and both are intimately associated with combustion, or stimulated by it; as the acids and minerals which are in solution in the water, and liberated by the heat, attack the boiler internally; whilst the sulphur which is liberated by the combustion of coal has a strong affinity

* See Roper's "Use and Abuse of Steam-Boilers."

for the iron of which boilers are constructed, and attack it externally.

Manual and Mechanical Firing.

The term firing is understood to be the art of applying fresh coal or other fuel to a furnace, which operation, in the case of large furnaces, incurs the severest kind of manual labor, and is attended with a great loss of fuel, in consequence of the great volume of cold air that enters the furnace every time the operation of replenishing or cleaning the fires is performed. Numerous attempts have been made to obviate this waste by the invention of machinery that would fire or supply the fuel continuously, but so far no mechanical arrangements have proved a success; nor is it at all likely that they ever will, as there are difficulties to be encountered which no human ingenuity in the design of machines can probably ever overcome. It is impossible to design a machine that will distribute the coal uniformly over the surface of the fire, including the sharp corners, etc. Unless that can be done, mechanical firing, however ingenious the arrangement may be, must ever prove a failure.

Even if a machine were devised that would distribute the fuel evenly over the fire-surface, it would not be available for cleaning the fires, and, as a result, there would be nearly the same loss incurred if the fires have to be cleaned by hand, as if they were fed by hand. This being the case, the question would naturally be asked, Why is it that thousands of dollars have been expended in attempts to fire mechanically? and the answer would be, that there are always parties to be found who are ready to devote time and invest money in every delusion which has ever been promulgated in connection with the steam-engine and boiler. If fuel could be consumed in round or oval furnaces, it would render more service than if burned in square furnaces, as there is always more or less dead material in the square corners through which the air escapes, thus lowering the temperature in the furnace, and rendering combustion less active and more wasteful.

Technical Terms applied to Firing.

Start Fires. — This term is understood to mean starting fresh fires in furnaces with shavings, wood, coal, etc.

Bank Fires. — This term is understood to mean covering the fires down with a thick body of coal at night, or when the engine has to be stopped for an indefinite period.

Slice Fires. — This means to push back the fire to the bridge-wall, and then draw out the cinders, after which the fire is drawn forward, distributed over the grates, and fresh fuel supplied. The terms slice and clean fires have the same meaning.

Draw Fires. — This term is understood to mean to draw the entire fire from the furnace for the purpose of allowing the furnace to cool for stoppage or repairs, as the case may be.

Technical Terms Employed in Relation to Boilers.

Curvilinear Seams. — The curvilinear seams of a boiler are those around the circumference.

Grate-Surface. — The term grate-surface means the aggregate number of square feet. In practice, the allowance of grate-surface is about three-fourths of a square foot per horse-power.

Longitudinal Seams. — The seams which are parallel to the length of a boiler are called the longitudinal seams.

Safe-working pressure, or safe load. — The safe-working pressure of steam-boilers is generally taken as $\frac{1}{5}$ of the bursting pressure, whatever that may be.

Steam-Room. — That part of a boiler occupied by the steam. In practice, it is about $\frac{1}{4}$ of the cubic contents of the boiler.

Water-Space. — That part of a steam-boiler which is occupied by the water. It is generally about $\frac{3}{4}$ of the cubic contents of the boiler.

The aggregate space in all classes of steam-boilers may be embraced under two heads, viz., steam-room and water-space.

Friction of Riveted Seams.

Owing to the contraction of rivets in cooling, the plates are, in many instances, brought into such close contact that the friction between them is sufficient to withstand the working strain without any shearing action coming upon the rivets. This is more especially the case with machine riveting. The contraction of a wrought-iron bar in cooling is nearly equal to $\frac{1}{10000}$ of its length for a decrease of temperature of fifteen degrees Fah., and the strain thus induced is about one ton for every square inch of sectional area in the bar.

Thus, if a rivet one inch in section were closed at a temperature of 900 degrees, it would in cooling decrease in length $\frac{60}{10000}$ of its length; and if its elasticity and strength remained perfect, would produce a tension of 60 tons. The ultimate strength of rivet iron, however, being only 24 tons, the rivet would in cooling be permanently elongated, and would continue, when cool, to exert a tension of 24 tons, providing its elasticity remained uninjured by the strain. Thus, if the rivets were not in contact with the plates, excepting at the head and tail, the plates would be held together by a pressure of 24 tons, and this friction would have to be overcome before the rivet came into action as a mere pin, from which will be seen that, by judicious riveting, the friction may, in many cases, be nearly sufficient to counterbalance the weakening of the plate from the punching of the holes.

Calking.

The object of calking is to bring together the seams of boilers, tanks, or hulls of iron vessels after riveting, so that they may be perfectly steam- or water-tight. This is done by using a sharp tool ground to a slight angle. The edge of the plates being first chipped or planed to an angle of about 110°, the calking-tool is applied to the lower edge of the chipped or planed angle, in order to drive or upset the edge, thus bringing the plates together, and

rendering the joint to all appearances perfectly steam-tight, and able to resist the internal pressure brought to bear upon this particular point. There are different methods of calking, but the concave method has many points of preference over any other. Boilers should never be calked while under steam- or water-pressure, however light, as the jarring induced by the calking is liable to spring the seams and cause fresh leakage in other parts of the boiler.

Steam-Boiler Explosions.

The principal causes of explosions,* in fact, the only causes, are deficiency of strength in the shell or other parts of the boilers, *over-pressure* and *over-heating.* Deficiency of strength in steam-boilers may be due to original defects, bad workmanship, deterioration from use or mismanagement. Deficiency of strength arising from bad workmanship is the most difficult to discover, and not unfrequently escapes the closest scrutiny, more particularly in the case of flue, tubular, and locomotive boilers.

Over-pressure may be caused by the safety-valve being overweighted; by its sticking on its seat; by the inadequate size of the communication between the boiler and valve, or by an incorrect and worthless steam-gauge. The same effect may be produced when there is a disproportion between the grate- and heating- surfaces, or where the heat from a large grate is concentrated on a small space. Under such circumstances, the heat is delivered with such intensity as to lift the water from the surface of the iron, thereby exposing it to the direct action of the fire.

Explosions occurring from excessive firing are in all cases the result of avarice, ignorance, or a want of skill in the care and management of the steam-boiler. Overheating may be caused by the accumulation of hard, solid incrustation adhering to the parts most exposed to the direct action of the fire, or it may be due to insufficiency of water, resulting from leakage of the valve

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* See Roper's "Use and Abuse of the Steam-Boiler."

or stop-cock, a failure in the supply-pipe, or a neglect to turn it on at the proper time or in sufficient quantity.

A steam-boiler may be well designed, of good material, and of first-class workmanship, and yet in a few months, after being put under steam, it may explode with terrible effect. On examining into the cause of the explosion, it may turn out that the water used made a heavy deposit; that the boiler had not been cleaned since it was put into use; that the fires had been fiercely urged, and the water driven from the surface of the iron; as a result, the life had been entirely burned out of the sheets over and around the fire, thereby weakening the boiler, and putting it in a dangerous condition. That the sudden heating or cooling, and oxidation of the boiler, induce great deterioration of strength has been proved by experience. Defects in the material, as blisters, lamination arising from inferior material, or want of care in the manufacture, are other sources of weakness in steam-boilers.

Safety-Valves.*

The safety-valve is designed on the assumption that it will rise from its seat under the statical pressure in the boiler, when this pressure exceeds the exterior pressure on the valve, and that it will remain off its seat sufficiently far to permit all the steam which the boiler can produce to escape around the edges of the valve. The problem then to be solved is: What amount of opening is necessary for the free escape of steam from a boiler under a given pressure? The area of a safety-valve is determined from formulæ based on the velocity of the flow of steam under different pressures, or experiments made to ascertain the area necessary for the escape of all the steam a boiler could produce under a given pressure. But as valves do not rise appreciably from their seats under varying pressures, the point to be considered is, how high any safety-valve will rise under the influence of a given pressure. This question cannot be determined theoretically, but has been settled conclusively by Burg, of Vienna, who ascertained from

* See page 654.

careful experiments that the rise of the valve diminishes rapidly as the pressure increases, as may be seen from the annexed table.

Pressure in Lbs.	Rise of Valve.	Pressure in Lbs.	Rise of Valve.
12	$\frac{1}{36}$	60	$\frac{1}{86}$
20	$\frac{1}{48}$	70	$\frac{1}{132}$
35	$\frac{1}{54}$	80	$\frac{1}{168}$
45	$\frac{1}{65}$	90	$\frac{1}{168}$
50	$\frac{1}{86}$		

In ordinary safety-valves, the average rise for pressures ranging from 10 to 40 pounds is about $\frac{1}{40}$ of an inch; from 40 to 70 pounds, about $\frac{1}{80}$, and from 70 to 90, about $\frac{1}{120}$ of an inch. The following table gives the result of a series of experiments made at the Novelty Iron Works, New York, some years ago, for the purpose of determining the exact area of opening necessary for safety-valves, per each square foot of heating-surface, at different boiler pressures.

Boiler Pressure in Lbs. Above the Atmosphere.	Area of Orifice in Sq. In. for Each Sq. Ft. of Heating-Surface.	Boiler Pressure in Lbs. Above the Atmosphere.	Area of Orifice in Sq. In. for Each Sq. Ft. of Heating-Surface.
0·25	·022794	40·	·001723
0·5	·021164	50·	·001389
1·	·018515	60·	·001176
2·	·014814	70·	·001015
3·	·012345	80·	·000892
4·	·010582	90·	·000796
5·	·009259	100·	·000719
10·	·005698	150·	·000481
20·	·003221	200·	·000364
30·	·002244		

Now, if we compare the area of openings, according to these experiments, with Zeuner's formula, which is entirely theoretical,

it will be observed that the results from the two sources are almost identical.

The lift of safety-valves, like all other puppet-valves, decreases as the pressure increases; but this seeming irregularity may be explained as follows: a cubic foot of water generated into steam at one pound pressure per square inch above the atmosphere will have a volume of about 1600 cubic feet. Steam at this pressure will flow into the atmosphere with a velocity of 482 feet per second. Now, suppose the steam was generated in five minutes, or in 300 seconds, and the area of an orifice to permit its escape as fast as it is generated be required, 1600 divided by 482 × 300 will give the area of the orifice, $1\frac{3}{5}$ square inches. If the same quantity of water be generated into steam, at a pressure of 50 pounds above the atmosphere, it will possess a volume of 440 cubic feet, and will flow into the atmosphere with a velocity of 1791 feet per second. The area of an orifice, to allow this steam to escape in the same time as in the first case, may be found by dividing 440 by 1791 × 300; the result will be $\frac{3}{25}$ square inches, or nearly $\frac{1}{8}$ of a square inch, the area required. It is evident from this that a much less lift of the same valve will suffice to discharge the same weight of steam under a high pressure than under a low one, because the steam, under a high pressure, not only possesses a reduced volume, but a greatly increased velocity; it is also obvious that a safety-valve, to discharge steam as fast as the boiler can generate it, should be proportioned for the lowest pressure.

There does not appear to be any recognized rule among boiler makers for proportioning safety-valves, since, while one allows one inch of area of safety-valve to every 66 square feet of heating-surface, another gives 1 inch area of safety-valve to every 4 horse-power, while a third allows 1 inch area of safety-valve to $1\frac{3}{4}$ square feet of grate-surface. This last proportion has been proved by experience to be capable of admitting of a free escape of steam, without allowing any greater increase of pressure than that for which the valve is loaded, providing that all the parts are in good working order. It is obvious, that no valve can act without a

slight increase of pressure, as, in order to lift at all, the internal pressure must exceed that of the load. Doubtless, most safety-valves are larger than is actually required, and but few boiler explosions occur from want of safety-valve area. The most probable causes of accidents arising from safety-valves are that they are either overloaded or out of order. A badly proportioned safety-valve, whether too large or too small, is objectionable, and is always attended with a certain amount of danger.

Rules.

Rule *for finding the weight necessary to put on a safety-valve lever, when the area of valve, pressure, etc., are known.*—Multiply the area of valve by the pressure in pounds per square inch; multiply this product by the distance of the valve from the fulcrum; multiply the weight of the lever by one-half its length (or its centre of gravity); then multiply the weight of valve and stem by their distance from the fulcrum; add these last two products together; subtract their sum from the first product, and divide the remainder by the length of the lever; the quotient will be the weight required.

Rule *for finding the pressure per square inch when the area of valve, weight of ball, etc., are known.*—Multiply the weight of ball by the length of lever, and multiply the weight of lever by one-half its length (or its centre of gravity); then multiply the weight of valve and stem by the distance from fulcrum. Add these three products together. This sum divided by the product of the area of the valve, and its distance from the fulcrum, will give the pressure in pounds per square inch.

Rule *for finding the pressure at which a safety-valve is weighted when the length of lever, weight of ball, etc., are known.*—Multiply the length of the lever in inches by the weight of the ball in pounds; then multiply the area of valve by its distance from the fulcrum; divide the former product by the latter; the quotient will be the pressure in pounds per square inch.

Rule *for finding centre of gravity of taper-levers for safety-valves.* — Divide the length of lever by two (2); then divide the length of lever by six (6); and multiply the latter quotient by width of large end of lever, less the width of small end, divided by width of large end of lever plus the width of small end. Subtract this product from the first quotient, and the remainder will be the distance in inches of the centre of gravity from large end of lever.

Dead-weight safety-valves are those in which a pressure is exerted on the valve by means of a weight suspended on the long arm of the lever.

Spring safety-valves are those in which the pressure of the steam against the face of the valve is resisted by means of a spiral spring. They are generally used for locomotives, as, in consequence of the jar, the dead-weight safety-valve is impracticable.

Lock safety-valves are those in which the weight on the lever is enclosed in a lock-box, to prevent the engineer from increasing the pressure at will. This arrangement of safety-valve is most generally used on the boilers of marine engines, tug-boats, and ferries.

Draught in Chimneys.

The presence of draught in any locality is due, to a certain extent, to the unbalanced pressure of the atmosphere, and is, in many cases, intensified and heightened by natural causes, but more frequently by mechanical and artificial arrangements. The natural draught or rush of air up chimneys or funnels is caused by the buoyancy both of the rarefied atmosphere and of the gases which pass through the fuel, as well as by the natural affinity of the colder and denser atmosphere to rush in and fill up the vacuum caused by the escape or ascension of the preceding volume. All the phenomena connected with draught are not as well understood as they should be, considering its importance as an agent in the promotion and maintenance of the combustion of fuel; the object of draught being to supply oxygen to the burning fuel, and disseminate or eject the products of combustion.

Numerous attempts have been made at different times to lay down rules for the area and height of chimneys that would produce sufficient draught for the consumption of a certain quantity of fuel in a given time, but such formulæ have more frequently failed, than succeeded, in giving satisfactory results, which is due probably to the want of knowledge of the requirements in each individual case, and of the location and surroundings. Attempts are, in many instances, made to produce a good draught by carrying the chimney above all surrounding objects and buildings, but it frequently occurs that shorter chimneys of the same area and internal dimensions have a better draught. It is claimed by some engineers that chimneys ought to increase in area from bottom to top, to be capable of producing a good draught, while others assert just the reverse, and claim that they ought to decrease from bottom to top. It has been found by experiment that both arrangements produced a good draught under some circumstances, but neither of them under all circumstances. The area of any chimney should increase slightly from bottom to top, in order to provide for the increased volume of the heated air and gases resulting from their expansion. It has been found that round flues produced a better draught, as a general thing, than either square or oval ones of the same area and height. This doubtless arises from the fact that air, rushing through or up a flue or funnel, has a tendency to assume the form of a screw, which is due probably to some natural cause.

Adverse currents and capping winds frequently interfere with the draught in short chimneys, but the same effect is frequently produced on tall ones during some kinds of weather and at certain seasons of the year; certain it is, that very tall stacks do not produce a corresponding draught in proportion to the height, and it has been demonstrated by observation that there is nothing to be gained by raising chimneys very high. It often occurs that chimneys of apparently sufficient height are incapable of producing sufficient draught. This, in many instances, arises from the fact that the quantity of fuel consumed in the furnace will not produce

sufficient heat in the flue to rarefy the air and cause draught, while in other chimneys of ample height and area, in consequence of the air and heated gases having to pass through a long, cold flue between the boiler and chimney, the draught is sluggish and unsatisfactory. There is no lack of formulæ for proportioning chimneys, which have been furnished by Wye Williams, Rankine, Weisbach, Trowbridge, Steel, Watt, and others, but each is only applicable in certain cases; and indeed it appears that Watt knew as much about proportioning the flue as any of our modern engineers, which may be inferred from the fact that modern writers on the subject refer to him as frequently as to any one else. This goes to show that we have not made such rapid advances in mechanical science, so far as regards proportioning chimneys to produce good draught under all circumstances, as might have been expected, considering the intelligence of the present generation and the progressive ingenuity of the age.

There are always individuals to be found who can tell how to proportion a chimney or a flue that will produce a draught sufficient to carry off the smoke and waste gases resulting from the consumption of a certain quantity of fuel, but they rarely ever explain all the conditions under which this may be accomplished; such as the distance between the furnace and chimney; whether the flue is perfectly straight, or contains a number of bends; and whether in its course it ascends or descends. Such information is akin to that which tells engineers that a pound of coal will evaporate 8 or 9 lbs. of water, but never gives the conditions under which it may be done, which include the type or design of boiler, the quality of the iron, the condition of the boiler for cleanliness, etc., the purity of the fuel, and the intelligence and experience of the care and management. It is well known to most experienced engineers that the boiler that will evaporate 9 lbs. of water per lb. of coal under some circumstances, will not evaporate over 5 lbs. of water per lb. of coal under others, and the results will be about the same in regard to draught.

A forced draught may be produced by various mechanical ar-

rangements, such as blowing-engines, fan-blowers, steam-jets, etc.; but, although it may be suitable, and even an absolute necessity in the prosecution of many branches of mechanical industries, a forced draught is objectionable in assisting the combustion of fuel for the generation of steam in ordinary steam-boilers, and never fails to induce mischievous effects, and consequently a good natural draught is very much to be preferred when attainable. Any flue ought to be as smooth on the inside as circumstances will permit, in order to diminish the friction between the walls of the flue and the escaping air and gases. And in regard to the height of chimneys and proportions of flues, it is always better to be governed by such practice as has given satisfaction in that locality, and with a particular kind of fuel, than to be guided by any theory, however scientific. The sectional area of the flue is what is termed the calorimeter of the boiler, and the calorimeter, divided by the length of the flue in feet, is termed the vent. The flues of all boilers diminish in their calorimeter as they approach the chimney, as the smoke contracts in volume in proportion as it passes through the heat.

Funnels.—The area of the funnels of steamships, tug-boats, and ferry-boats varies considerably with different builders and in different countries. The number of circular inches per nominal horse-power is given in the following table, for several makers.

Highest,	15·14	Highest,	12·96	Highest,	14·45	Highest,	16·40	Highest,	14·06
Mean,	14·10	Mean,	11·79	Mean,	13·94	Mean,	15·94	Mean,	13·12
Low,	13·01	Low,	10·89	Low,	12·96	Low,	15·14	Low,	12·17

Mean Total, 13·78

These are all for low pressures. For high pressure, the number of inches varies from 9·11 to 6·02, mean 7·07. The funnel should evidently bear a proportion to the amount of heated air and smoke passing through it, which must bear a nearer proportion to the horse-power than to the surface of the fire-grate. Where the fire-grate is small, a large quantity must be burned per square foot. If, in one case, 20 lbs. of coal are burned per square foot per hour, and in another 40 lbs., and the funnels are propor-

tioned to the fire-grate, they will not be proportioned to their requirements.

Rule *for finding the required area for the chimneys of stationary boilers.*— Multiply the nominal horse-power of the boiler by 112, and divide the product by the square root of the height of the chimney in feet. The quotient will be the required area in square inches.

A well-proportioned and moderately high smoke-stack is to be preferred for sea-going steam-vessels, as tall ones are difficult to steady on account of the oscillation of the vessel, arising from the disturbance of the water and the resistance of the wind.

Superheaters.— Superheaters are steam-chambers located in the uptakes of marine-boilers or at the base of the funnel, and so arranged that the waste heat from the furnaces may pass around and through them, prior to escaping up the chimney. They are used for drying the steam in its transit from the boilers to the steam-cylinders of the engines. The heat or flame passes through the tubes and around the shell, the steam being inside. They are fitted with a stop-valve, and arrangements for mixing the superheated and saturated steam, or using either independently; they also have safety-valves similar to those used on steam-boilers. There is no definite size for superheaters, as they are not intended for a receptacle for any large amount of steam, but simply as a means of drying it. The proportionate area of superheating to heating surface in modern marine-boilers is about 1 to 10 square feet.

An intercepter or separator is a chamber attached to marine-boilers for the purpose of intercepting the water carried out by the steam. The steam enters at the top and strikes against a partition plate, then passes under it and escapes to the cylinder; the water which enters with the steam is collected in the bottom of the box and drawn off through a valve.

Smoke.

Smoke once formed in a furnace, flue, or chimney can never be burned by any mechanical device or arrangement, nor can there

be any advantage in incurring much expense in the attempt, except to abate a nuisance, as very little economy in fuel would result from the adoption of any such device. A very general idea prevails that, when we see large volumes of smoke issuing from the mouths of the chimneys of stationary boilers, smoke-stacks of locomotives, and funnels of marine-boilers, whenever fresh fuel has been applied, a great waste of fuel is taking place; this, however, is a mistake, as about $\frac{7}{10}$ of the volume is steam resulting from the moisture expelled from the coal, wood, or shavings by the application of heat; besides, sulphur and other earthy matters which, like the steam, are incombustible, enter into and increase the volume.

This may be easily explained by stating that $\frac{1}{2}$ ton of water is converted into steam in the furnace for every ton of bituminous coal consumed, which is an actual benefit, because, if the carbon had not been thoroughly mixed with such a great mass of steam, it would have fallen in the shape of a black cloud of dust in the locality where the furnace was situated, and have become a more insufferable nuisance than the smoke. Smoke contains about 20 per cent. of combustible and 80 per cent. of incombustible matter. Such being the case, the question would naturally arise, Would it be advisable to incur much expense in an attempt to consume 80 per cent. of incombustible matter, for the purpose of gaining 20 per cent.?

Feed-Water Heaters.

The benefits to be derived from heating the feed-water for boilers by exhaust steam may be explained as follows: A pound of feed-water entering a steam-boiler at a temperature of 50° Fah., and evaporated into steam of 60 lbs. pressure per square inch, requires as much heat as would raise 1157 pounds of water 1 degree. A pound of feed-water raised from 50° Fah. to 220° Fah. requires 987 thermal units of heat, which, if absorbed from exhaust steam passing through a heater, would be a saving of 15 per cent. in fuel. Feed-water, at a temperature of 200° Fah., entering a boiler,

as compared in point of economy with feed-water at 50°, would effect a saving of over 13 per cent. in fuel; and with a well constructed heater there ought to be no trouble in raising the feed-water to a temperature of nearly 212° Fah.

If we take the normal temperature of the feed-water at 60°, the temperature of the heated water at 212°, and the boiler-pressure at 20 lbs., the total heat imparted to the steam in one case is $1192{\cdot}5° - 60° = 1132{\cdot}5°$, and in the other case $1192{\cdot}5° - 212° = 980{\cdot}5°$, the difference being 152°, or a saving in fuel of $\frac{152}{1132{\cdot}5} = 13{\cdot}4$ per cent. Supposing the feed-water to enter the boiler at a temperature of 32° Fah., each pound of water will require about 1200 units of heat to convert it into steam, so that the boiler will evaporate between $6\frac{3}{4}$ and $7\frac{1}{2}$ pounds of water per pound of coal. The amount of heat required to convert a pound of water into steam varies with the pressure, as will be seen by the following table:

TABLE

SHOWING THE UNITS OF HEAT REQUIRED TO CONVERT ONE POUND OF WATER, AT THE TEMPERATURE OF 32°, INTO STEAM AT DIFFERENT PRESSURES.

Pressure of Steam in Lbs. per square inch by Gauge.	Units of Heat.	Pressure of Steam in Lbs. per square inch by Gauge.	Units of Heat.
1	1·148	110	1·187
10	1·155	120	1·189
20	1·161	130	1·190
30	1·165	140	1·192
40	1·169	150	1·193
50	1·173	160	1·195
60	1·176	170	1·196
70	1·178	180	1·198
80	1·181	190	1·199
90	1·183	200	1·200
100	1·185		

If the feed-water has any temperature, the heat necessary to convert it into steam can easily be computed. Suppose that its temperature is 65°, and that it is to be converted into steam having a pressure of 80 lbs. per square inch, the difference between 65 and 32 is 33; subtracting this from 1181 (the number of units of heat required for feed-water having a temperature of 32°), the remainder, 1148, is the number of units for feed-water with the given temperature.

Technical Terms applied to Adjuncts of the Steam-Boiler.

Angle-irons.—Irons used for the purpose of staying steam-boilers. See page 400.

Air-casing.—An arrangement attached to fire- and smoke-box doors for the purpose of preventing radiation of heat.

Blast-pipe.—A small pipe used to blow steam into the funnels of marine-boilers for the purpose of exciting the draught in the furnace.

Blow-off cocks.—Cocks used for blowing the water out of steam-boilers.

Check-valve.—A valve used to retain the water in steam-boilers, and relieve the feed apparatus from the pressure.

Check-chamber.—The chamber in which the check-valve operates.

Connecting-pipes.—The pipes which connect check-valves with steam-boilers.

Crown-sheet.—That part of fire-box boilers (locomotive or marine) directly over the fire.

Crown-bars.—Bars placed on the upper side of crown-sheets, in the water-space, for the purpose of strengthening them.

Crown-braces.—Braces attached to the crown-bars, and to the shells and domes of boilers, for the purpose of resisting the pressure exerted on the flat surfaces of crown-sheets.

Dashers.—Iron plates which are sometimes attached to the inside of steam-boilers to prevent the cold water, as it enters, from striking the tubes.

Dead-plate.—The solid iron plate which fills the space between the end of the grate-bars and the fire-door of boiler-furnaces.

Deflector.—An arrangement employed, in the furnaces of locomotives and marine-boilers, for the purpose of mixing the air and gases arising from the combustion of the fuel, and causing them to ignite.

Diaphragm-plate.—A perforated plate, used in the steam-domes of locomotives and marine-boilers, to prevent the water from being carried over into the cylinder with the steam.

Dome.—An elevated chamber on the top of steam-boilers, from which the steam is generally taken for the cylinders.

Dome-stays.—Stays employed, in the domes of locomotives and marine-boilers, for the purpose of strengthening them.

Gasket.—A packing employed for making the man- and hand-holes of steam-boilers steam- and water-tight.

Gauge-cocks.—Cocks used on the front-head of steam-boilers by which to ascertain the height of the water.

Grummet.—A packing of hemp, used between the flanges of steam- and water-pipes, for the purpose of making them steam- and water-tight.

Stay-tubes.—Tubes used for bracing marine-boilers. They are generally made of thicker material than either the ordinary fire- or water-tubes.

Spanner-guard.—An arrangement employed to secure cocks and valves, connected with marine-engines and boilers, from being opened or closed by accident.

Scum-cocks.—Cocks employed to blow off extraneous substances from the surface of the water in steam-boilers.

Spectacles.—Pieces of iron, with concave sides, employed as braces between the tubes of marine-boilers, generally for the purpose of stopping leaks.

Tube-sheets.—The sheets into which the tubes are inserted at each end of the boiler.

Knees.—Brackets riveted to the sides of steam-boilers, for the purpose of sustaining them on their supports.

Waist.—A term applied to the cylindrical part of locomotive- or marine-boilers.

Instructions for the Care and Management of Steam-Boilers.

On first entering a boiler-room in the morning, ascertain whether the water stands at the proper level or not.

Never start a fire under a boiler until you are satisfied there is sufficient water in it.

On taking charge of an engine and boiler, first ascertain if there is sufficient water in the boiler, and then trace out the pipes and connections between the engine, boiler, and pumps.

In starting a fresh fire under a boiler while it is cold, always allow it to burn gradually at first, in order to bring all the parts of the boiler to a uniform temperature.

Never blow out a boiler under a head of steam, as the heat remaining in the boiler will bake the scale and mud on the sheets and flues, after which it will be impossible to soften it again.

When preparing to clean boilers, allow them to cool down, and the water to remain in them until ready to commence cleaning.

Never fill a boiler with cold water while the shell, flues, or tubes are hot, as the contraction induced by the tube in cooling will have an injurious effect.

Boilers, under which a forced draught is used, require to be cleaned oftener than when the draught is natural.

Never carry a higher pressure of steam than is necessary, nor allow the water to rise above the second gauge-cock in the boiler when the engine is running.

Before starting a fire under a boiler, place a small quantity of coal on the grates, to prevent them from being warped by the extra heat of the new fire.

Boilers should be cleaned and examined inside and out every three months.

Never neglect to blow out and clean boilers, even although solvents are used for the prevention and removal of scale.

Never put a new boiler into service until examined thoroughly for the purpose of ascertaining if the boiler-makers have neglected to remove all lamps, hammers, tools, etc.

Never open a steam-valve, on a boiler under pressure, quickly, for the purpose of allowing steam to escape into the atmosphere, or into a boiler containing a less pressure, as it is attended with a certain amount of danger, and may possibly produce an explosion.

Clean the flues or tubes of the boiler at least once a week, and never allow ashes or cinders to accumulate under the grates.

Never throw water around the furnaces of fire-box boilers.

If the water should, from any unforeseen cause, become dangerously low, draw the fire, allow the boiler to cool down, and neither admit feed-water nor disturb the safety-valve.

In case the supply of water should be temporarily cut off, owing to the derangement of a pump, the bursting of a pipe, or any other cause, stop the engine, cover the fire with fresh coal, and shut the

damper, so as to retain a sufficient quantity of water in the boiler to start on.

When it becomes necessary to blow out a certain quantity of the water from a boiler every day, the hand should never be removed from the cock or valve, as any diversion of a person's attention from it may allow too much to be blown out, and the boiler be ruined.

In all cases where it is possible, regulate the feed-water so as to send it into the boiler in a steady stream.

When fresh water is used in marine-boilers, it is best to use salt water for a short time when first put into use, in order to cover the parts with a thin coat of scale. This prevents them from being injured by the action of fresh water.

The term salting marine-boilers, means that the flues, tubes, and crown are covered with a thick coating of salt, which prevents the water from coming in contact with the iron. This induces cracking and burning of the parts so coated, besides causing a great waste of fuel.

The parts of marine-boilers most likely to suffer from an insufficiency of water are the tubes and crowns; but the water cannot become low in marine-boilers from accident, as they can be fed either from the boiler feed-pumps, circulating, independent, donkey, or bilge pumps.

If a tube becomes leaky in the tube-sheet, it may be made tight by inserting a tapering iron ferule about $\frac{1}{16}$ of an inch larger than the inside diameter of the tube.

If a tube splits, it may be plugged with either iron or wooden plugs, whichever is most convenient. Iron is best for the end next the furnace, while wood will answer for the smoke-box end.

Boiler Materials.

Boiler making now holds an important place among the mechanical arts. Its progress has been aided chiefly by the enormous growth of the steam-engine as the prime mover, by the increased

facilities afforded for procuring suitable materials, and by the improvements made in working them. In the early days of the steam-engine, boilers of copper and cast-iron were used for generating steam, but they were seldom subjected to a pressure higher than that of the atmosphere; but when pressures of 3 to 4 or even 7 atmospheres came into use, cast-iron was found to be unreliable and treacherous, for which reason it was discarded in favor of *wrought iron*, which was not employed at first, in consequence of the difficulty found in working it and in making steam-tight joints. It has, however, of late years become the material employed to the almost entire exclusion of all others. It has been more extensively employed in the construction of steam-boilers, for the past thirty years, than any other material, on account of its great tensile strength, its ductility, power of bearing sudden and trying strains, trustworthy nature, the ease with which it can be welded, riveted, patched, or mended, and its moderate first cost, etc.

The first quality to be sought for in boiler materials is strength. This does not necessarily imply the mere power to resist being torn asunder by a dead weight, as in a testing-machine; but the quality to withstand, without injury, the varying shocks and strains to which boilers are exposed. An inferior quality of plates cannot be relied upon to bear the ordeal of heating and cooling repeatedly, as they invariably warp and twist, showing defects of manufacture; more especially in the process of cold bending, when minute fractures often occur on the outer surface of the plates of stubborn or inferior qualities of iron.

The defect most commonly revealed in working boiler-plates is want of lamination. This defect arises from the imperfect welding of the several layers which make up the thickness of the plate, and is usually caused by interposing sand or cinder, which has not been expelled by hammering or rolled out during the process of manufacture. This is more frequent in thick than in thin plates, and is sometimes very difficult to detect in cold plate, although often discernible in the hot. It also often happens that plates which are passed as quite sound, on careful external examination

are found to be severely laminated when subjected to heating and hammering, and prove totally unfit for use.

Blisters are of a similar nature, and arise from the same cause as lamination. Sometimes they appear as mere surface defects, and are of no consequence; but their appearance may be an indication of want of care or skill in the making of the plate, and should always excite suspicion. It frequently happens that these defects pass undetected after the closest scrutiny and test by hammering, but disclose themselves soon after the boiler is set to work, especially if the plates be exposed to sudden variations of temperature. In the plates over the fire-grate of an externally fired boiler, such a blister may prove a very serious defect, and often necessitates the cutting out and replacement of the sheet. Inferior brands of iron will rapidly show unmistakable signs of weakness when placed under the trying ordeal of bearing the alternate impingement of a fierce flame and currents of cold air. The rapid variations of temperature caused by the sudden and frequent openings of the furnace door, and passage of cold air through the grate-bars, will soon tell on even the best iron, but more quickly on that of an inferior brand.

Characteristics of boiler-iron when broken. On breaking a plate or bar of wrought-iron, the fracture presents an appearance by which the quality of the iron may, in some measure, be determined. The fracture is designated, on the one hand, as fibrous, tough, silky, close-grained, etc., or, on the other hand, crystalline, coarse, open-grained, brittle, and cold-shut. When broken suddenly, the best qualities of plate and bar iron exhibit a fine, close-grained, uniform crystalline fracture, even silky, of a light silver color; the appearance in the harder descriptions approaching to that of steel. The appearance of indifferently refined and inferior qualities is coarser, usually of a darker color, more or less uneven, or open, exhibiting large facets, and approaching some descriptions of cast-iron. When broken gradually, good iron presents a well drawn out, close fibre, of light greenish hue, whilst inferior qualities give a shorter, more open, and darker fibre.

When good ductile iron is gradually torn asunder, it stretches to a considerable extent, causing a diminution of sectional area at the fractured part, which should always be compared with the original sectional area of the specimen in judging of the quality. An inferior bar or plate may bear as great a tensile strain as a similar specimen of superior quality; but on comparing their fractured areas, it will generally appear that the latter has been drawn out considerably, whilst the inferior specimen, having stretched but little, has not sensibly diminished at the fracture. This is owing to the fact that good ductile iron, when sudden strains occur, will stretch, while badly refined will snap. Wrought-iron changes from fibrous to crystalline, after enduring long-continued cold hammering, vibration, tension, jarring, and other strains, after long exposure to the influence of heat, or alternate expansion and contraction whenever it has been used for the plates of a boiler furnace. Even the very best plates, after from ten to twenty years' use in a boiler, have frequently been found to break without stretching, at the same time displaying a crystalline fracture.

It has been said that this shows that a change has taken place in the nature of the material, and that, from being fibrous and tough, it has, by some unexplained cause, become crystallized and brittle, or that it has lost its nature in consequence of the treatment it has undergone, whatever that may have been. There is no doubt that the strains and other causes above mentioned have a tendency to make good iron become brittle and liable to snap suddenly under the same treatment that would originally have torn it gradually, and to this extent a change is produced in its nature. This snapping, and not the fatigue of the metal, is the direct cause of the crystalline fracture, which is but a necessary consequence of the suddenness of the breaking, and not a property of the iron itself. To say it snaps readily because it has become crystalline is to confound the cause with the effect. It is erroneous to say the fibrous nature has passed out of the iron, as its ductility can to some extent, at least, be restored, in most cases, by simply heating to a bright red, and slowly cooling, the

iron, or, failing that, by hammering or rolling it while hot. By heating to redness, and suddenly cooling, a piece of wrought-iron, it will become liable to snap, producing the same effect as cold hammering. The explanation of this is not clear, and it may be owing to the loosening of the crystals into which the composition of the material ultimately resolves itself. To this cause may also be attributed the same tendency to snap after long-continued jarring or alternate expansion and contraction.

It may be asserted, without fear of contradiction, that all boiler-plate worthy of the name is fibrous; whether its hardness makes it liable to snap, and, therefore, appear crystalline, depends on its original character and the treatment it has undergone. No fine iron can, however, by any treatment, except burning, be made to appear coarse, and the fibres of the poorest descriptions of iron cannot, without refining, be made to appear fine and close-grained. From a want of knowledge of the above facts, false opinions are often expressed respecting the qualities of boiler-plates.

It is no unusual thing to find intelligent mechanics and boiler-makers expressing their opinions, at coroners' inquests, on the quality of the iron in exploded boilers, without anything to base their opinions on except the load per square inch required to tear the plates asunder. They seem to forget, if the boiler be an old one, that the age, the position in the boiler in which the rent has taken place, the amount of strain to which it has been exposed, and all the circumstances connected with the occurrence, should be known in order to decide understandingly as to the quality of the iron. It has been shown, in numerous instances, that good ductile iron can be made to appear crystalline when pulled asunder in the testing-machine, by confining the minimum sectional area where fracture will occur to one point or to a very short length.

The general conclusions with regard to boiler material, which may be regarded as established from experiments, observations, and practice, thus far seem to be, 1st, That the laws of resistance of the parts of boilers to the internal pressure are sufficiently well established; 2d, It is of the utmost importance that the ma-

terials employed should be of the best quality as regards strength and durability; and as there are but few manufacturers of boiler-plates, the inspection of materials, especially boiler-plates, should be made by competent persons, appointed for that purpose, at the place of manufacture, which inspection should extend to the qualities of ores and the process of manufacture, the required brands, stamps, or certificates being put on or authorized by the inspectors in person. There is much greater certainty of securing the best materials by an inspection of the process of working, and of the raw materials employed, than by an inspection of plates after they have been sent to market, when, judging from all external appearances, good and bad plates are not easily distinguished.

Practical limits to the thickness of boiler-plates.—The proper strength of boilers, in order to enable them to withstand with safety the required pressure of the steam, is a matter of much importance as regards both life and property, and the responsibility of the proprietors and constructors of boilers is of so grave a character as to justify the devotion of a much larger space to this subject than is convenient in this work. The principles on which the strength of the material depends may be expressed in a very few words,—the strength being directly as the thickness of the metal, and, inversely, as the diameter of the boiler.

So long as the quality of boiler-iron remains as it is at present, the thickness of the plate may be practically determined within exceedingly narrow limits, as a good boiler must be constructed of plate ranging in thickness from $\frac{1}{4}$ to $\frac{1}{2}$ an inch, as anything less than the former cannot be properly caulked, and any thickness greater than the latter is difficult to rivet without the aid of machinery. A thickness of $\frac{3}{8}$ seems to have become the standard thickness for all diameters of boilers intended to sustain a high pressure. This, perhaps, arises from the fact that boiler-makers seem to be better acquainted with the practical limit to the strength of that thickness, because it has of late years been used more than any other; nevertheless, for steel, or some of the higher grades of American plate, a less thickness will suffice for the same pressure.

Definitions of the Technical Terms Applied to the Different Kinds of Boiler-Plate.

C. No. 1 charcoal iron means that charcoal was the fuel employed in the blast-furnace when the iron was smelted. Such iron is not suitable for any purpose when exposed to a high temperature. Although it is frequently used for the shells of boilers, it is very seldom employed for furnace-sheets.

C. H. No. 1 charcoal iron, commonly called flange-iron, is manufactured by the same process as C. No. 1, with this difference, that it is reheated and hammered, which increases its compactness, solidity, and strength as well as its capacity for resisting high temperatures. C. H. No. 1 is generally called cold blast-iron; the process of manufacture is as follows. The pig-metal is remelted and refined, or converted into wrought-iron in charcoal fires, the balls being hammered into blooms. These blooms are reheated in reverberatory furnaces, and then rolled into slabs. These pieces are called covers, between two of which clippings of boiler-plate and other wrought-iron scraps are placed, after which the mass is brought to welding heat and passed between heavy rollers. The greatest danger to be encountered in this process arises from the imperfect welding of the pieces. It is often due to the slag which remains between the coils when the mass is heated. Iron manufactured by this process frequently blisters when exposed to an intense heat. Boiler-plate should never be manufactured by this process, as it is generally of inferior quality, and always proves deceptive. The only advantage in making it in this manner is cheapness. C. H. No. 1 charcoal iron is produced by piling one slab upon another at right angles with each other, and exposing them to a high welding heat, after which they are rolled and hammered, great care being taken both in the selection of the material and in the rolling and hammering.

Fire-box iron is a kind of plate manufactured exclusively for furnaces. It is produced in the same manner as C. H. No. 1, with this difference, that it is subjected to two or three more processes

of heating, rolling, and hammering. There are many grades of this kind of iron, resulting from the details of the processes which are customary in the different plate mills, and the care with which the iron is selected. The names of the different manufacturers furnish a better guarantee than the stock of knowledge possessed by the most talented experts.

Iron produced from covers filled with iron scrap will blister, unless the slag is expelled by patient and careful heating, rolling, and hammering. Such iron, if used for fire-box plates, should be tested as follows. Lay the plate off with a straight-edge, and pencil or chalk in squares of about one foot each; then, with a light steel hammer, strike the surface of each square about one inch apart, when, if there are any defects in the iron, they will in all probability be made manifest by the sound. As soon as each square is finished, it should be cancelled, in order to prevent repetition. If the iron is perfect, it will give out a clear sound.

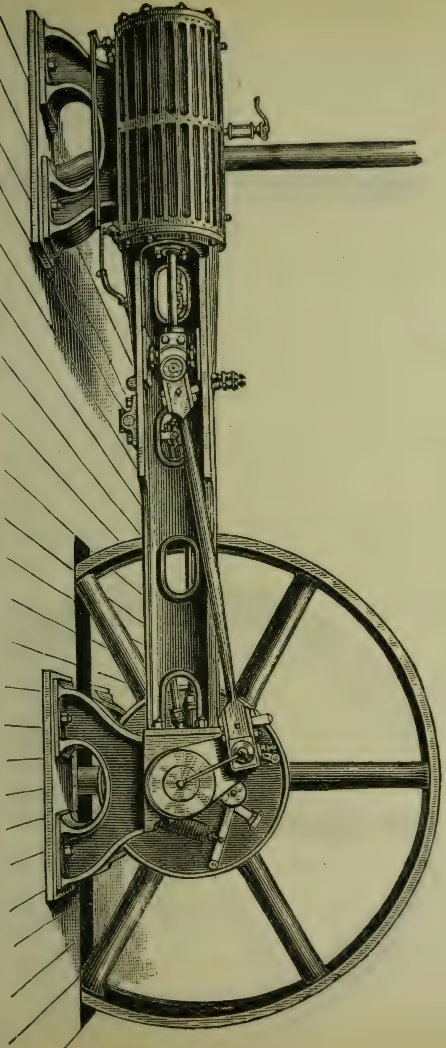
From the foregoing, it will be seen how much depends on the character of the material, and the care taken in the process of manufacture. It is well known that in many instances the iron in different plate mills is the same in every respect, and, while the processes through which it has passed are the same to all appearance, on examination it has been found that that produced by one mill was of an excellent quality, while that produced by another was of a very inferior grade. As a general rule, boiler-plate that can be bent at right angles, when heated to a red heat, without showing any cracks, may be relied upon. But the indications of superiority will be strengthened, if the iron can stand the test of bending at right angles when cold, as none but the finest grades can bear it.

Steel boiler-plates are generally made of puddled steel, in which the ordinary puddling process, by means of which wrought-iron is made from pig-iron, is arrested at the point required for the carbonization of the steel. Homogeneous steel plates are produced from cast-steel, which is formed by melting the finest grades of wrought-iron in crucibles with carbonaceous matter, after which the ingots are reheated and rolled into plates of the desired thickness.

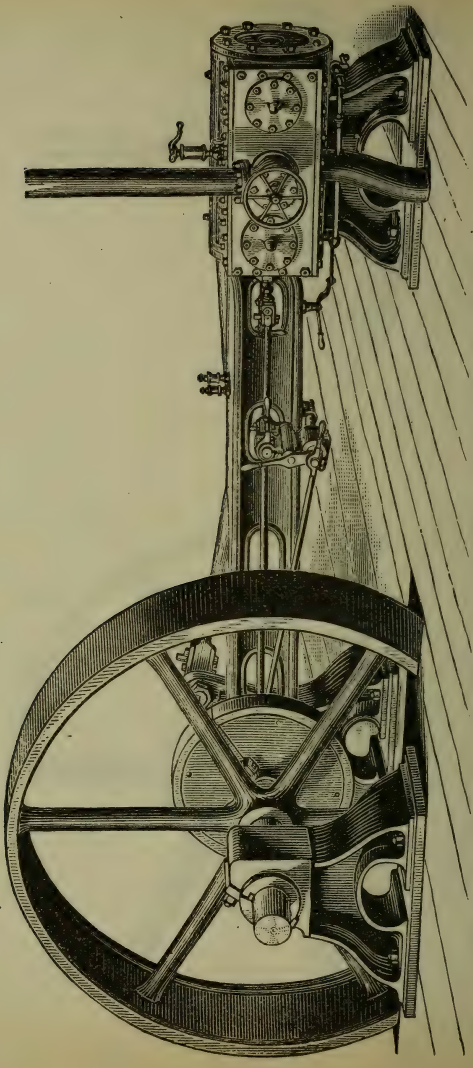
The Buckeye Automatic Cut-Off Engine.

The cuts on pages 489, 490, represent a front and back view of the Buckeye automatic cut-off steam-engine. As may be observed, the bed-plate is a modification of the Corliss or girder-frame pattern, a design which possesses sufficient rigidity, without extra weight of metal. It is faced up at one end to receive the cylinder, and at the other the main pillow-block. The cylinder contains the steam-ports, but not the exhaust-ports; and, as the valve-faces are as near the cylinder as is consistent with sufficient strength, the clearance is reduced to a minimum, a feature which renders the engine very economical in the use of steam. The cross-head is made in halves, is held together by bolts, and is attached to the piston-rod by means of a thread on the rod. The cross-head shoes move in flat guides, and can be easily adjusted by means of screws and jam-nuts.

The main steam-valve is driven by a fixed eccentric in the usual manner. An adjustable eccentric, the position of which on the shaft is under the control of the governor, works the cut-off valves. A novel feature of the cut-off valve-gear is a rock-shaft working in a bearing in the rocker-arm belonging to the main valve-gears. The adjustable eccentric is attached to a pendant arm on the outer end of it, and a similar but vertical arm on the inner end connects it to the head, and thus works the cut-off valve. The effect of this device is to secure a correct movement of the cut-off valves relatively to their seats in the moving main valve, and at the same time to effect a degree of adjustment of the cut-off exactly corresponding to the degree of change in the angular position of the eccentric, neither of which is possible without such an arrangement. These engines are in very general use, and are said to be very durable and economical. One of them on exhibition at the Centennial Exposition at Philadelphia attracted considerable attention. They are manufactured (both condensing and non-condensing) by the Buckeye Engine Company, Salem, Ohio, under J. W. Thompson's patent.



Front View of the Buckeye Automatic High-Pressure Cut-Off Engine.



Back View of the Buckeye Automatic High-Pressure Cut-Off Engine.

Questions,

THE ANSWERS TO WHICH MAY BE FOUND IN THE TEXT.

Define the term steam-boiler.

Why is there more need of accurate information in relation to the steam-boiler than to the steam-engine?

What causes affect the strength and durability of steam-boilers?

What qualities are most desirable in a steam-boiler?

Describe the nature and effect of the destructive forces, both chemical and mechanical, that act on steam-boilers.

Of what form should a boiler be constructed to embody the greatest strength?

Give the names of the different boilers in use, both land and marine, their advantages and disadvantages.

State the proportion of heating-surface to grate-surface sufficient to constitute a horse-power in a steam-boiler.

What conditions will influence the amount of water which one pound of coal will evaporate in a steam-boiler, also the maximum and minimum evaporation per pound of coal?

Give the principal causes which induce foaming in steam-boilers.

Give the names of the different adjuncts of steam-boilers.

Give the rule for finding the bursting-pressure of steam-boilers.

Give the rule for finding the safe working-pressure of steam-boilers.

Give the rule for finding the internal strain caused by the pressure of steam on the shells of steam-boilers.

Give the rule for finding the pressure per square inch of sectional area on the crown-sheets of steam-boilers.

Give the rule for finding the safe external pressure of boiler-flues.

Give the rule for finding the collapsing-pressure for boiler-flues.

Give the rule for finding the number of square feet of heating-surface in any given number of flues or tubes.

Give the rule for finding the relative strength of single- and double-riveted seams of steam-boilers.

Give the rule for finding the strength of stays for steam-boilers.

Give the rule for finding the heating-surface for any steam-boiler.

Explain the object of stay-bolts, their breaking-strength, etc.

Explain the causes which induce the formation of scale in steam-boilers.

Give the chemical ingredients of the scale which forms in steam-boilers.

Explain the causes of the loss of fuel induced by incrustation in steam-boilers.

What are the causes which induce deterioration in steam-boilers?

Does the tensile strength of boiler-iron increase by the application of heat? and, if so, up to what degree Fah. does it increase?

What are the causes of corrosion in steam-boilers? and what analogy does corrosion bear to combustion?

What advantage has mechanical firing over manual firing, and *vice versâ*?

Give the technical terms as applied to firing.

Give the technical terms employed in relation to steam-boilers.

Explain the cause of friction in riveted seams.

What is the object of caulking?

Explain the causes of steam-boiler explosions.

What is the object of a safety-valve on a steam-boiler?

Give the rule for finding the weight necessary to be placed on a safety-valve lever when the area of the valve, pressure, etc., are given.

Give the rule for finding the pressure per square inch against the safety-valve when the area of the valve, weight of ball, etc., are known.

Give the rule for finding the pressure at which the safety-valve is weighted when the length of the lever, the weight of the ball, etc., are known.

Give the rule for finding the centre of gravity of taper levers of safety-valves.

Explain the comparative advantages and disadvantages of dead-weight, spring, and lock safety-valves.

Explain the cause of draught in chimneys.

Explain the advantages and disadvantages of square, oval, and circular chimneys.

Give the rule for finding the area of a chimney or funnel necessary to produce a sufficient draught to consume a given quantity of fuel in a given time.

What are the advantages of superheaters?

What is the object of an interceptor?

What are the chemical ingredients which constitute smoke?

Can smoke, when once formed, be consumed by any mechanical process?

Does the formation of smoke incur a waste of fuel, and, if so, to what extent?

Explain the meaning of the technical terms applied to the different adjuncts of steam-boilers.

What course should an engineer or fireman pursue when first entering the boiler-room in the morning?

What precaution should be taken before starting a fire under a boiler?

What course should an engineer adopt on taking charge of an engine and boiler for the first time?

How should the fire be regulated when first started under a boiler?

Under what conditions should a boiler be blown out?

What should be the condition of a boiler when it is to be filled with cold water?

What course should be adopted with boilers before cleaning?

How should boilers be treated when a *forced draught* is used?

How should the pressure in a boiler be regulated?

How should the kindling material be placed on the grate preparatory to starting a fire?

How often should steam-boilers be cleaned?

Should the cleaning of boilers be neglected, when solvents are used for the prevention and removal of scale?

What precautions should be taken before new boilers are put into service?

How often should the flues or tubes of boilers be cleaned?

What course should be adopted in case the water in a boiler becomes dangerously low?

What course should be pursued in case the water-supply should become interrupted for any length of time?

What precaution should an engineer take, in case it becomes necessary to blow out a certain quantity of water every day?

How should the supply of feed-water be regulated?

What advantages are gained by filling marine-boilers with salt-water for the first time?

What is the meaning of the term "salting" when applied to marine-boilers?

What parts of any class of steam-boilers are most likely to suffer from the effects of heat?

What is the most practical method to adopt in case a boiler-tube should become leaky?

What course should an engineer or fireman adopt in case a tube should become split?

Give the characteristics of good boiler material, whether iron, steel, or copper.

Give the definitions of the technical terms applied to the different kinds of boiler-plates.

PART SEVENTH.

Air.

The atmosphere is known to extend at least 45 miles above the earth. Its aggregate weight has been calculated at upwards of 77,000,000,000 of tons, or equivalent to the weight of a solid globe of lead 60 miles in diameter. Hence, this enormous weight reposes incessantly upon the earth's surface, and upon every object, animate or inanimate, solid, liquid, or aëriform. 100 cubic inches of air at the surface of the earth, when the barometer stands at 34 inches, and at a temperature of 60° Fah., weigh about 31 grains, being thus about 815 times lighter than water, and 11,065 times lighter than mercury. The component parts of the air are about 79 measures of nitrogen gas and 21 of oxygen; or, in other words, air consists of (by volume) oxygen, 21 parts; nitrogen, 79 parts (by weight); oxygen, 77 parts; nitrogen, 23 parts.

Now, since the air is possessed of weight, it must be evident that a cubic foot of air at the surface of the earth has to support the weight of all the air directly above it; and that, therefore, the higher we ascend in the atmosphere, the lighter will be the cubic foot of air; or, in other words, the farther from the surface of the earth the less will be the density of the air. At the height of three and a half miles, it is known that the atmospheric air is only half as dense as it is at the surface of the earth. From the nature of fluids, it follows that the atmosphere presses against any body with which it comes in contact — because fluids exert a pressure in all directions — upwards, downwards, sidewise, and obliquely. Its particles are so inconceivably minute, that they enter all substances, even liquids. It penetrates all the ramifications and innermost recesses of porous bodies, and is mixed up with and circulates in the blood of men and animals; and by the pressure of its superincumbent strata, it is urged through almost every substance. It

is this circulation through the interior of the bodies of men and animals which counterbalances its outer pressure; because, if its weight were not neutralized, neither man nor beast could walk, and would be as mute as statues of lead, and lips once closed could never again be opened.

The amount of pressure of a column of air, whose base is one square foot and whose altitude is the height of the atmosphere, has been found to be 2156 pounds avoirdupois, or very nearly 15 pounds of pressure on every square inch. Consequently, it is common to state the pressure of the atmosphere as equal to 15 pounds on the square inch. If any other gaseous body or vapor — such as steam — exerts a pressure equivalent to 15 pounds on the square inch, then the force of that vapor is said to be equal to one atmosphere. If the vapor be equal to 30 pounds on every square inch, then it is equal to two atmospheres, and so on; consequently, the atmospheric pressure is capable of supporting about 30 inches of mercury, or a column of water 34 feet high.

It is known that the pressure of the atmosphere is not constant, even at the same place. At the equator, the pressure is nearly constant, but is subject to great changes in high latitudes. In some countries the pressure of the atmosphere varies so much as to support a column of mercury so low as 28 inches, and at other times so high as 31, the mean being 29·5; thus making the average pressure between 14 and 15 pounds on the square inch. But in scientific books, generally, the pressure is understood, in round numbers, to be 15 pounds; so that a pressure exerted equal to 1, 2, 3, 4, etc., atmospheres means such a pressure as would support 30, 60, 90, 120, etc., inches in a perpendicular column, or 15, 30, 45, 60, etc., pounds on every square inch.

The pressure of the air differs at different altitudes; * at 7 miles above the surface of the earth, the air is four times lighter than it is at the surface; at 14 miles it is 16 times lighter; and at 21 miles it is 64 times lighter. It requires 13,817 cubic feet of air

* See table on page 498.

to make one pound; consequently, one cubic foot of air at the surface of the earth weighs 527 grains, or ¼ of an ounce avoirdupois; but under a pressure of 5½ tons to the square inch, air becomes as dense, and would weigh as much per cubic foot, as water.

TABLE

OF ALTITUDES ABOVE SEA-LEVEL, AND THE CORRESPONDING ATMOSPHERIC PRESSURES, DEDUCED FROM THE OBSERVATIONS OF THE HAYDEN EXPEDITION TO THE ROCKY MOUNTAINS.

Location.	Altitude in Feet.	Pressure of the Atmosphere.
Altoona, Pa.	1,168·	14·08
Cairo, Ill.	291·23	14·56
Cheyenne, Wy. Ter.	6,075·28	11·48
Cincinnati, O.	440·	14·46
Cresson, Pa.	2,000·	13·64
Denver, Col.	5,196·58	11·94
Golden City, Col.	5,728·98	11·67
Lake Champlain	100·84	14·64
" Erie	573·08	14·39
" Huron	589·99	14·38
" Michigan.	589·15	14·39
" Ontario.	249·99	14·56
Louisville, Ky.	404·	14·48
Mt. Lincoln, Col.	14,296·66	7·06
New Albany, Ind.	379·75	14·5
Ogden, Utah	4,303·3	12·42
Omaha, Neb.	977·9	14·18
Pike's Peak, Col.	14,148·66	7·1
Pittsburg, Pa.	699·2	14·33
Rock Island, Ill.	566·68	14·40
St. Louis, Mo.	429·29	14·17
Terre Haute, Ind.	485·55	14·44

TABLE

SHOWING THE FORCE OF THE WIND IN POUNDS PER SQUARE FOOT AT DIFFERENT VELOCITIES.

Miles per Hour.	Feet per Second.	Force per Square Foot Pound.	
1	1·47	0·005	Hardly perceptible.
2	2·93	0·020	Just perceptible.
3	4·4	0·044	
4	5·87	0·079	Gentle, pleasant wind.
5	7·33	0·123	
6	8·8	0·177	
7	10·25	0·241	
8	11·75	0·315	Pleasant, brisk gales.
9	13·2	0·400	
10	14·67	0·492	
12	17·6	0·708	
14	20·5	0·964	
15	22·00	1·107	
16	23·45	1·25	
18	26·4	1·55	Very brisk.
20	29·34	1·968	
25	36·67	3·075	
30	44·01	4·429	High wind.
35	51·34	6·027	
40	58·68	7·873	
45	66·01	9·963	Very high.
50	73·35	12·30	
55	80·7	14·9	
60	88·02	17·71	Storm or tempest.
65	95·4	20·85	
70	102·5	24·1	Great storm.
75	110·	27·7	Hurricane.
80	117·36	31·49	
100	146·66	50·	Tornado.

Horse-Power of Wind Storms.

It is asserted that severe wind storms exert a pressure of from 25 to 30 lbs. per square foot, and travel from 50 to 70 miles per

hour. Assuming that the pressure is 30 lbs. per square foot, or $\frac{1}{5}$ of a pound per square inch, with a speed of 66 miles per hour, then, as there are 27,878,400 square feet, or 4,014,489,600 square inches in a square mile, if the pressure of the storm was exerted for the height of half a mile, it will give an area of 2,007,244,800 square inches for each mile in width upon which the storm acts.

Rule *for finding the horse-power of wind storms.*

Multiply the area acted on in inches by the pressure in lbs. per square inch; then multiply this product by the speed in feet per minute, and divide by 33,000. The quotient will be the horse-power of the storm.

Example. — 2,007,244,800 square inches × 1·5 lbs. pressure × 5800 feet ÷ 33,000, which gives as a result 70,557,700 horse-power developed for each mile of breadth of the track of the storm. To produce the same horse-power with improved engines consuming but two pounds of coal per hour per horse-power, would require 63,000 gross tons of coal.

Altitude of the Highest Mountains in the World.

The highest peak of the Himalayas, in Asia, is 25,659 feet above sea-level.

Mont Blanc, the highest peak of the Alps, is 15,732 feet.

The highest peak of the Andes is 14,760 feet.

The peak of Teneriffe is 11,454 feet.

Mount Ætna is 9,000 feet.

The highest point in the Pyrenees is 8,400 feet.

The highest inhabitable point on the globe is Ancomarsa, one of the Peruvian Andes, which is 16,000 feet.

Highest Waterfalls in the World.

The Ribbon Falls, Yosemite Valley, U. S. A., 3,300 feet.

Yosemite Falls, U. S. A., 2,600 feet.

The Arve Falls, Bavaria, Europe, 2,000 feet.

The Falls of Montmorency, Canada, 250 feet.

Niagara Falls, United States, 158 feet.

TABLE

SHOWING THE RELATIVE VOLUMES OF AIR AT VARIOUS TEMPERATURES.

Temp. Fah.	Volume in Cubic In.	Temp. Fah.	Volume in Cubic In.	Temp. Fah.	Volume in Cubic In.	Temp. Fah.	Volume in Cubic In.
—49	834·7	— 6	922·5	37	1010·2	80	1098·0
—48	836·7	— 5	924·5	38	1012·2	81	1099·0
—47	838·8	— 4	926·5	39	1014·3	82	1100·0
—46	840·8	— 3	928·6	40	1016·3	83	1102·1
—45	842·8	— 2	930·6	41	1018·4	84	1104·1
—44	844·9	— 1	932·7	42	1020·4	85	1106·2
—43	846·9	— 0	934·7	43	1022·4	86	1108·2
—42	849·0	1	936·7	44	1024·5	87	1110·2
—41	851·0	2	938·8	45	1026·5	88	1112·3
—40	853·1	3	940·8	46	1028·6	89	1114·3
—39	855·1	4	942·9	47	1030·6	90	1116·4
—38	857·1	5	944·9	48	1032·7	91	1118·4
—37	859·2	6	947·0	49	1034·7	92	1120·4
—36	861·2	7	949·0	50	1036·7	93	1122·5
—35	863·3	8	951·0	51	1038·8	94	1126·5
—34	865·3	9	953·1	52	1040·8	95	1128·6
—33	867·3	10	955·1	53	1042·9	96	1130·6
—32	869·4	11	957·1	54	1044·9	97	1132·7
—31	871·4	12	959·2	55	1046·9	98	1134·7
—30	873·5	13	961·2	56	1049·0	99	1136·7
—29	875·5	14	963·3	57	1051·0	100	1138·8
—28	877·6	15	965·3	58	1053·1	101	1140·8
—27	879·6	16	967·3	59	1055·1	102	1142·9
—26	881·6	17	969·4	60	1057·1	103	1144·9
—25	883·7	18	971·4	61	1059·2	104	1147·0
—24	885·7	19	973·5	62	1061·2	105	1149·0
—23	887·8	20	975·5	63	1063·3	106	1151·0
—22	889·8	21	977·6	64	1065·3	107	1153·1
—21	891·8	22	979·6	65	1067·3	108	1155·1
—20	893·9	23	981·6	66	1069·4	109	1157·1
—19	895·9	24	983·7	67	1071·4	110	1159·2
—18	898·0	25	985·7	68	1073·5	111	1161·2
—17	900·0	26	987·8	69	1075·5	112	1163·3
—16	902·0	27	989·8	70	1077·6	113	1165·3
—15	904·1	28	991·8	71	1079·6	114	1167·3
—14	906·1	29	993·9	72	1081·6	115	1169·4
—13	908·2	30	995·9	73	1083·7	116	1171·4
—12	910·2	31	998·0	74	1085·7	117	1173·5
—11	912·2	32	1000·0	75	1087·8	118	1175·5
—10	914·3	33	1000·2	76	1089·8	119	1177·6
— 9	916·3	34	1004·1	77	1091·8	120	1179·6
— 8	918·4	35	1006·1	78	1093·9	121	1181·6
— 7	920·4	36	1008·2	79	1095·9	122	1183·7

TABLE — (*Continued.*)

Temp. Fah.	Volume in Cubic In.	Temp. Fah.	Volume in Cubic In.	Temp. Fah.	Volume in Cubic In.	Temp. Fah.	Volume in Cubic In.
123	1185·7	152	1244·9	180	1302·0	208	1359·2
124	1187·8	153	1246·9	181	1304·1	209	1361·2
125	1189·8	154	1249·0	182	1306·1	210	1363·3
126	1191·8	155	1251·0	183	1308·2	211	1365·3
127	1193·9	156	1253·0	184	1310·2	212	1367·3
128	1195·9	157	1255·1	185	1312·2	213	1369·4
129	1198·0	158	1257·1	186	1314·3	214	1371·4
130	1200·0	159	1259·2	187	1316·3	215	1373·5
131	1202·0	160	1261·2	188	1318·4	216	1375·5
132	1204·1	161	1263·3	189	1320·4	217	1377·5
133	1206·1	162	1265·3	190	1322·4	218	1379·6
134	1208·2	163	1267·3	191	1324·5	219	1381·6
135	1210·2	164	1269·4	192	1326·5	220	1383·7
136	1212·2	165	1271·4	193	1328·6	230	1404·1
137	1214·3	166	1273·5	194	1330·6	240	1424·5
138	1216·3	167	1275·5	195	1332·6	250	1444·9
139	1218·4	168	1277·5	196	1334·7	260	1465·3
140	1220·4	169	1279·6	197	1336·7	270	1485·7
141	1222·4	170	1281·6	198	1338·8	280	1506·1
142	1224·5	171	1283·7	199	1340·8	290	1526·5
143	1226·5	172	1285·7	200	1342·9	350	1546·9
144	1228·6	173	1287·8	201	1344·9	400	1751·0
145	1230·6	174	1289·8	202	1346·9	500	1955·1
146	1232·7	175	1291·8	203	1349·0	600	2159·2
147	1234·7	176	1293·9	204	1351·0	700	2363·3
148	1236·7	177	1295·9	205	1353·1	800	2567·4
149	1238·8	178	1298·0	206	1355·1	900	2771·5
150	1240·8	179	1300·0	207	1357·1	1000	2975·6
151	1242·9						

Technical Terms which are Applied to Fluids and Vapors, and which Bear a Certain Relation to the Steam-Engine.

Vaporization.—Vaporization is the act or process of vaporizing liquids, or converting them into vapor.

Diffusion of vapor.— Diffusion of vapor means the state of being scattered, as steam on escaping from the mouth of an exhaust-pipe is wafted away and scattered over a great extent of space.

Compressibility means the quality of being compressible or being capable of being compressed into a smaller space, while incompressibility implies the opposite property.

Conductibility means the quality of being conductible, that is, of being capable of being conducted or conveyed away.

Expansion means the state of being expanded or being capable of expanding, either in surface or bulk.

Boiling-point means the temperature at which fresh water will boil at sea-level, which is generally understood to be 212° Fah.

Ebullition is the motion produced in a liquid by the rapid conversion of a part of it into vapor by the application of heat.

Condensation means the process of converting vapors into fluids by the abstraction of a portion of their heat mechanically.

Evaporation is a term applied to all bodies existing in an aëriform state; while spontaneous evaporation means the natural tendency inherent in all fluids to evaporate.

Fuel.

The word fuel is used to denote substances which may be burned by means of atmospheric air with sufficient rapidity to evolve heat capable of being applied to economical purposes. Fuel consists either of vegetable matter or of the products of the natural or artificial decomposition of such matter. Vegetable matter, which consists principally of woody tissue, is composed of carbon, hydrogen and oxygen, comprising the organic part, and a small proportion of so-called earthy matter, that which is inorganic. The sun is the source of the heat-producing power of fuel, since the organic parts are derived from water, and, except in particular cases, from the carbonic acid of the atmosphere, which are decomposed in the economy of plants by the action of solar light.

Hydrogen in fuel must always be in association with carbon, but carbon practically free from hydrogen may be procured abundantly and applied as fuel. In all fuel containing carbon, hydrogen, and oxygen, the proportion of hydrogen may be equal to or greater, but never less, than that required to form water with the oxygen. It is only the hydrogen in excess of this which is available as a source of heat, so that, in the combustion of a substance whose composition is represented by carbon and water, the carbon alone is the source of heat. The hydrogen existing in combination with oxygen in the state of water, so far from contributing to the actual amount of heat produced, must be evaporated at the expense of the heat developed by the combustion of the carbon.

If we compare different fuels, and assign them a value for heating purposes based on their chemical constitution, we will find that petroleum is about 25 per cent. superior to all others theoretically; in round numbers, it is capable of evaporating 15 lbs. of water per pound of fuel, while a pound of anthracite coal can evaporate 11 lbs., and a pound of coke only about 9 lbs.; these figures varying, to a certain extent, with the different qualities of the fuels.

The chemical properties of coal are, free carbon, hydro-carbons, water or oxygen, and hydrogen, with solid matter termed ash; the proportions of these vary considerably. In some instances, the solid matter is 25 per cent., while with superior coal, only 6 or 10 per cent. The products of combustion are carbonic acid gas, nitrogen, air, ashes, and steam.

The oxygen necessary for the combustion of coal is derived from the atmosphere. One pound of carbon in combustion unites with 2·66 lbs. of oxygen, and the product is 3·66 lbs. of carbonic acid gas. From the above it will be seen that to the 2·66 lbs. of oxygen 11 lbs. of air would have to be brought into contact with the pound of coal (if pure carbon) to render its combustion complete; but, as coal contains hydrogen, it is found that instead of 11, 12 lbs. are required.

The value of wood as fuel compared with coal. — Two and a

half pounds of dry wood are equal to one pound (average quality) of soft coal, and the fuel value of the same weight of different woods is very nearly the same, — that is, a pound of hickory is worth no more for fuel than a pound of pine, assuming both to be dry. If the value be measured by the weight, it is important that the wood be dry, as each 10 per cent. of moisture or water in the wood will detract about 12 per cent. from its value as a fuel.

The weight of one cord of different woods (air-dried) is as follows:

Hickory, or Hard Maple	4500 lbs.
White Oak	3850 "
Beech, Red Oak, and Black Oak	3250 "
Poplar, Chestnut, and Elm	2350 "
Pine	2000 "

The fuel value of wood, as compared with coal, is about as follows:

1 Cord air-dried Hickory, or Hard Maple, equal to	2000 lbs. coal.
1 Cord air-dried White Oak equal to	1725 " "
1 Cord air-dried Beech, Red Oak, or Black Oak equal to	1450 " "
1 Cord air-dried Poplar, Chestnut, or Elm equal to	1050 " "
1 Cord air-dried Average of Pine Wood equal to	925 " "

Comparative value of different kinds of wood for fuel.

Shellbark Hickory	100	Yellow Oak	60
Pignut Hickory	95	Hard Maple	59
White Oak	84	White Elm	58
White Ash	77	Red Cedar	56
Dog-Wood	75	Wild Cherry	55
Scrub Oak	73	Yellow Pine	54
White Hazel	72	Chestnut	52
Apple-Tree	70	Yellow Poplar	51
Red Oak	67	Butternut and White Birch	43
White Beech	65		
Black Birch	62	White Pine	30

Fire. — Fire is one of the oldest chemical phenomena. Its discovery was one of the greatest boons conferred on mankind, as with it arose sociability, the family joys of the domestic hearth, all industries and arts, together with the wonders they have produced, and still produce from day to day. Hence, we can readily understand how it is that fire has ever been, and still is, among nations the object of a special worship (priests of Baal, Gebers, Hindoos, Brahmans, etc.), and has often figured in the religious or funereal rites of nations most remote from each other, both in time and space, as the Chaldees, Hebrews, Greeks, Romans, Peruvians, Mexicans, etc. But how and when this great discovery was made, in the absence of which we can hardly conceive of the possibility of human arts, or even of human existence, is unknown.

Flame. — Flame is gas or vapor, of which the surface, in contact with the atmospheric air, or other supporter of combustion, burns with the emission of light. The luminosity of flame is generally admitted to be caused by the presence of particles of solid matter within, or in immediate contact with, the gas in active combustion.

Smoke. — Smoke is the product of imperfect combustion, caused either by a want of oxygen or a want of temperature. Bituminous coal contains from 5 to 6 per cent. of hydrogen, which unites with the oxygen necessary to combustion, and constitutes water. A ton of bituminous coal will make nearly one-third of a ton of water in the form of steam. That this steam is black, does not necessarily indicate the presence of much carbon, as a grain of soot, if distributed evenly in fine particles through a cubic foot of steam, would color it blacker than the ace of spades.

Chemical analysis proves the basis of soft coal to be carburetted hydrogen, but it generally contains benzole, naphtha, asphaltum, paraffine, lubricating oil, and a great variety of other substances used in the mechanical arts.

Heat.

According to the dynamical or mechanical theory, heat is the result of motion among the atoms of matter, or, as it may be otherwise stated, of inter-atomic movement; and this motion is capable of being propagated through space, from one body to another, by undulations of a so-called ether assumed to be everywhere existent in the universe.

The relative effect of such heat producing motion, or, in other words, the relative proportions of heat required to cause given effects, may be accurately indicated by numbers, just as if heat were a ponderable agent; and it is usual to speak of heat as if it were an independent material substance: thus, it is said to be evolved, or emitted, radiated, conducted, absorbed, and stored up, or accumulated. As a variable amount of the heat evolved in the combustion of a body is absorbed in the work of effecting alterations in the physical condition of the combustible elements necessary to their effective oxidation, it is impossible to estimate the absolute quantity of heat evolved by the combustion of a body; yet the relative quantities of heat evolved by the combustion of different bodies which may be utilized, can be accurately determined.

One of the remarkable effects of the application of heat to matter is, that the same amount will affect equal weights of dissimilar kinds in different degrees. Thus, the amount of heat that will raise 1 lb. of water from 100° to 200° Fah, will raise 30 lbs. of mercury through the same range. The amount that will raise 1 lb. of water 1°, will raise 14 lbs. of air.

The capacity of a body for heat is termed its *specific heat*, and may be defined as the number of units of heat necessary to raise the temperature of 1 lb. of that body 1° Fah.

The thermal unit, or unit of heat, as it is termed, is the quantity of heat that will raise 1 lb. of pure water 1° Fah., or from 39° to 40° Fah.

The term *latent heat* means the quantity of heat which has dis-

appeared from a body, owing to an increase of temperature. The sensible heat is that which is sensible to the touch or measurable by the thermometer.

The mechanical equivalent of heat is the amount of work performed by the conversion of one unit of heat into work, and the *mechanical theory* of heat is based on the assumption that heat and work are mutually convertible.

TABLE

SHOWING THE LATENT HEAT OF VARIOUS SUBSTANCES.

	Fah.		Fah.
Ice	140°	Steam	990°
Sulphur	144	Vinegar	875
Lead	162	Ammonia	860
Beeswax	176	Alcohol	442
Zinc	493	Ether	301

TABLE

SHOWING THE RADIATING PROPERTIES OF DIFFERENT SUBSTANCES.

	Fah.		Fah.
Water	100°	Blackened Tin	100°
Lampblack	100	Clean Tin	12
Writing-Paper	100	Scraped Tin	16
Glass	90	Ice	85
India-Ink	88	Mercury	20
Bright Lead	19	Polished Iron	15
Silver	12	Copper	12

TABLE

SHOWING THE EFFECTS OF HEAT UPON DIFFERENT BODIES.

	Fah.		Fah
Cast Iron thoroughly smelted	2,754°	Lead melts at	594°
Fine Gold melts at	1,983	Bismuth "	476
Fine Silver "	1,850	Tin "	421
Copper "	2,160	Tin and Bismuth, equal parts } melt at	283
Brass "	1,900	Alcohol boils at	174
Zinc "	740	Ether "	98
Quicksilver boils at	630	Mercury melts at	39
Linseed Oil "	600		

TABLE

SHOWING THE SPECIFIC HEAT OF DIFFERENT SUBSTANCES.

SOLIDS.

Copper	0·0951	Brass	0·0939
Gold	0·0324	Glass	0·1977
Iron	0·1138	Ice	0·5040
Lead	0·0314	Sulphur	0·2020
Platinum	0·0324	Charcoal	0·2410
Silver	0·0570	Alumina	0·1970
Tin	0·0562	Stones, Bricks, etc., about	0·2200
Zinc	0·0955		

LIQUIDS.

Water	1·0000	Mercury	0·0332
Lead (melted)	0·0402	Alcohol	0·6150
Sulphur "	0·2340	Fusel Oil	0·5640
Bismuth "	0·0363	Benzine	0·4500
Tin "	0·0637	Ether	0 5034

TABLE

SHOWING THE RELATIVE WEIGHT AND VOLUME OF DIFFERENT GASES.

Air	0·238	0·169
Oxygen	0·218	0·156
Hydrogen	3·405	2·410
Steam Gas	0·480	0·346
Carbonic Acid Gas	0·217	....
Nitrogen	0·244	....
Olefiant Gas	0·404	0·173
Carbonic Oxide	0·245	0·237
Ammonia	0·508	0·299

TABLE

SHOWING THE NON-CONDUCTING PROPERTIES OF DIFFERENT MATERIALS AT EVEN THICKNESS.

Black Slate	100
Sandstone	71·95
Fire-Brick	61·70
Soft Chalk	56
Asphaltum	45
Oak Wood	33·66
Pine Wood	27·61
Wood and Plaster	25·55
Sulphate of Lime	20·26
Sulphate of Lime and Sand	18·70
Coarse Ashes, Shavings, Hay, and Straw	25–85
Sawdust and Tan-Bark (fine)	17–20
Mineral Wood or Asbestos, cemented	18–20
Fine Asbestos, in thread	13–15
Fine-Powdered Charcoal	14–16
Ordinary Mineral Wool, Hair-Felt, Cat-Tail, etc.	10–13
Extra Mineral Wool, Raw Silk, Cotton, etc., quite loose	8–10
Ice	0

Cooling of liquids and solids.—The velocities with which a solid body cools in a liquid are approximately the same, whether it be placed near the surface or near the bottom. It is slightly less when the body is brought immediately under the surface. The nature of the external surface of the cooling body has but little influence. The velocity of cooling increases very considerably for the same body immersed in the same liquid with increasing temperature of the latter. If the cooling power of water be taken at 1, that of alcohol is equal to 0·58; mercury, 2·07; sulphate of copper, 1·03, and common salt, 1·05.

Combustion.

Combustion is a subject of interest to the engineer, manufacturer, and individual, and must ever continue to be so, while the

steam-engine is used as a motive power, and so long as artificial heat is employed for manufacturing and domestic purposes, as well as for the preservation of animal life. This subject has not heretofore received that consideration which its importance in an economical point of view so eminently deserves. This arose, in part, from the lavish hand with which a bountiful Nature has supplied us with minerals, woods, and cereals, and the close proximity of the source of supply to the avenue of demand; but the increase of population and demand, and the diminution in supply, are making the examination of the subject an imperative necessity. It is quite common to see in the neighborhood of manufacturing establishments, and even households, splendid lumps of the finest qualities of anthracite coal, nearly pure carbon, lying in the highway, to be forced into the ground by the pressure of hoof or wheel, and after rain storms dumping-grounds glisten with kernels of coal that have never been exposed to the fire, which are as fine as, and in many respects superior to, that which has been placed in the furnace. The same thing may be said of oil, cotton-waste, piston-rod packing, etc.; but, as the cost of material has to be paid out of the profits of production, such carelessness is generally followed by retributive justice; and the old adage which says " that a wilful waste is generally followed by a woeful want," is sooner or later realized.

Combustion is the result of chemical alterations of a violent character, and the heat thus evolved is merely an incidental phenomenon, or a vehement combination of various materials. In combustion, the carbon and oxygen have so great a chemical affinity for each other, that they rush violently together, and by the force of their combustion produce instant heat.

The composition of anthracite coal of the best quality is as follows: carbon, 90·45; hydrogen, 2·43; oxygen, 2·45, and ashes 4·67, with a minute quantity of nitrogen. When coal is heated, it discharges its gas; the solid carbon then ignites in presence of oxygen, and retains the temperature necessary for combustion as long as the necessary quantity of oxygen is applied. The average

weight of anthracite coal is about 53 lbs. per cubic foot, and the number of cubic feet per ton will average about 42·3.

Bituminous coal is a compound substance. A ton (2000 lbs.) contains about 1600 lbs. or 80 per cent. of carbon; 100 lbs. or 5 per cent. of hydrogen; and 300 lbs. or 15 per cent. of oxygen, nitrogen, sulphur, and ashes. The weight of bituminous coal will average about 50 lbs. per cubic foot and 44·8 cubic feet to the ton, and in the process of coking it loses 35 per cent. of its original weight.

TABLE

SHOWING THE TOTAL HEAT OF COMBUSTION OF VARIOUS FUELS.

Sort of Fuel.	Equivalent in Pure Carbon.	Lbs. of Water Evaporated from 212° Fah.	Lbs. of Water Raised 1° Fah.
Anthracite coal .	1·05	15·75	15225
Bituminous " .	1·06	15·90	15370
Coke	0·94	14·00	13620
Charcoal . . .	0·93	14·00	13500
Dry wood. . .	0·50	7·50	7000

Spontaneous combustion.—This mysterious phenomenon has attracted at different times the attention of chemists and philosophers, and many theories have been advanced to account for its development. Galletly, who investigated the subject, found that cotton-waste soaked in boiled linseed-oil, and wrung out, if exposed to a temperature of 170°, set up oxidation so rapidly as to cause actual combustion in 105 minutes. Coleman also instituted a very extensive series of experiments upon fragments of cotton, linen, jute, and woollen waste saturated with oils of different natures.

The theory which attributes spontaneous combustion to the presence of pyrites in the coal, may partially account for the increased number of fires; but Richter has shown that, for various coals experimented upon, those which contained the most

pyrites were not the most subject to spontaneous combustion. According to him, air is rapidly absorbed by the coal, and the oxygen of the air then combines with the organic components to produce carbonic acid and develop heat. According to all probabilities, however, the heat which determined the spontaneous combustion is due both to the oxidation of the iron and to that of the carbonized matters. This confined in badly-ventilated holds speedily reaches a temperature sufficiently high to produce combustion.

That most of the bituminous coals (English and American) are subject to spontaneous combustion when in bulk, and under favorable circumstances, has long been known. Experiments by Greenamann have also proved conclusively that an exposure of bituminous coal in heaps to the action of the weather for a period varying from two weeks to a year results in a large percentage of loss. This loss is in the nature of a slow or incomplete combustion; it is greater and more rapid in large heaps than in small, and is also favored by the greater or less state of subdivision of the coal, large fragments losing proportionably less than smaller ones. The loss varies from 5 to 25 per cent.

The higher the temperature the more rapid is the combustion. The heat around the coal-bunkers of steamships must necessarily be very great, from their close proximity to the boilers and furnaces; and in sailing-ships containing large quantities of these coals in bulk, taken on board mostly wet, the generation of heat to the point of ignition seems to be only a question of time. The sulphur and volatile matter in bituminous and hydrogenous coals are the active agents in spontaneous combustion, and the finer the particles the more favorable is the condition for producing that result. The large number of disasters, which have occurred from the spontaneous combustion of bituminous coals on board of steamships and sailing-vessels, has called public attention to the matter. Although the manner by which bituminous coal stored in vessels becomes ignited is not yet determined, it has been demonstrated that the conditions for the work of spontaneous combustion exist

wherever large bodies of bituminous coal are stored in close compartments.

From the foregoing considerations, it would seem that, when spontaneous combustion takes place among coals or other substances, drowning out with water is not always effective; as, though it extinguishes the fire, it leaves in the coal a condition of things very favorable to a renewed ignition at any moment. A terrible explosion of coal-gas recently occurred on board of a steamship in Liverpool, by which fourteen men were injured, some of them seriously, in consequence of a quantity of wet coal having been placed in the bunkers and the hatches closed.

Water.

Water, with the barometer at 30°, boils in the open air, at sea-level, at 212° Fah.; and in vacuum, at 88° Fah. The less the pressure of the atmosphere, the lower is the temperature at which water will boil. The pressure of the atmosphere at sea-level is 14·7 lbs. per square inch, pressing equally and in all directions. This has been ascertained from the following illustration. Because the height of a column of air of one square inch area exactly balances a column of mercury of the same area 30 inches in height, and also a column of water 33·86 feet in height, it follows that a column of air, 30 inches of mercury, and 33·86 feet of water weigh the same, and since the last two weigh respectively 14·7 lbs. per square inch, a full column of air must weigh the same. A cubic foot of water evaporated under a pressure of one atmosphere, or 15 lbs. per square inch, occupies a space of 1700 cubic feet.

Salt water boils at a higher temperature than fresh, owing to its greater density, and because the boiling-point of water is increased by any substance that enters into chemical combination with it. Mud and other substances, so long as they are kept in mechanical solution, will not increase the boiling-point of water; when these substances settle, and burn to the interior of the boilers, the boil-

ing-point will be increased. The density of water decreases as the temperature increases, since heat destroys cohesion and expands the particles, causing them to occupy greater space. The power of water to hold chemical substances, such as salts of lime, in solution, decreases as the temperature increases; from this it follows that boilers carrying high-pressure steam form more scale than those working at low temperatures.

The law of expansion by heat and contraction by cold is true as relating to water, with this exception, that, as hot water cools down from the boiling-point, it contracts until 45° Fah. is reached, but if cooled down from this point it expands again. The density of water decreases as the temperature increases, because water is expanded into a greater space by an increase of temperature. The cohesive attraction of the particles is not so great, and the water is therefore less buoyant, thus allowing the hydrometer to sink lower than it should.

Water, like all liquids, expands by the application of heat, and this fact alone shows the fallacy of the commonly accepted notion that it is incompressible; the dilation and contraction of the liquid is simply extension and compression of its particles. Although the expansion of water is comparatively slight between its boiling and freezing points, yet it is the most irregular of all liquids; so irregular, in fact, that it has been found impossible to find a single empirical formula to express the expansion at different temperatures. Below 50° Fah. it is more irregular than above that point, as water possesses what no other liquid has been discovered to have, and that is a point of maximum density.

If we take a water thermometer and expose it to the cold, we shall observe the following curious phenomenon. The liquid will gradually descend until it reaches the temperature of 39·2° Fah.; at this point the contraction will cease; and, although the cold acting on the bulb is far below this point, the liquid will gradually ascend until it reaches 32° Fah., or freezing point, when it will solidify. The point at which the liquid commences to ascend is called its "point of maximum density."

One of the most curious phenomena connected with water before and after freezing, may be demonstrated as follows: Take a tall jar and fill it with water, say at 60° Fah.; at the top of the jar fix a small mercurial thermometer, and another one at the bottom; then place the jar at rest, exposed to the cold. The lower thermometer will be observed to fall more rapidly than the top one, until it reaches 39·2° Fah., when it will remain stationary. The top thermometer will now fall, and continue to do so until the water freezes; the bottom thermometer still remaining at 39·2° Fah. These effects are easily explained: the particles of water at the top being exposed to the cold, decrease in temperature, thus becoming denser, and fall to the bottom, their places being taken up by warmer particles, which in their turn undergo the same change, until the whole volume has completely circulated, and attained a temperature of 39·2° Fah. The particles now, instead of becoming denser, actually expand, and so remain at the top until a thin layer of ice is formed. This is exactly what takes place in our lakes and ponds during every frost; the circulation continues until the whole mass attains the temperature of 39·2° Fah., when it is gradually and finally arrested; a thin layer of ice is then formed at the top, acting as a cloak to the interior, which, remaining always at 39·2° Fah., preserves the animals and fishes from the action of intense cold.

Were it not for this fact, our lakes and rivers would all be frozen at the bottom, and, as water is a bad conductor of heat, they would in time be converted into a solid block of ice, which would defy the hottest rays of a tropical sun to melt. Thus we see that such a wise provision of Nature depends entirely on an apparent exception to a universal law, which is so slight that it requires the most delicate experiments to detect it. The freezing point of a liquid is almost invariably the same as its melting point; that is, if we cool a liquid below its melting point, it will become solid. There are, of course, many exceptions to this, and even water has been known to be cooled down to 4° Fah. without freezing. To effect this, however, water most be kept perfectly

still, as, with the least vibration, congelation commences, and the temperature will instantly rise to zero.

When a substance solidifies or freezes, there is always a change of volume, which usually is a contraction; but, in the case of water, an expansion takes place. The expansion of water at the freezing point is by no means gradual, but takes place almost instantaneously, and the amount of force exerted at the time is enormous. It has been demonstrated by actual experiments, that in freezing, water exerts a pressure of about 30,000 lbs. per square inch, which far surpasses the strain that any of our machinery could bear.

Pure water is composed of hydrogen and oxygen in the proportions of two measures of hydrogen to one of oxygen, or one part of hydrogen to 8 of oxygen; or oxygen, 89 parts by weight, and by measure 1 part; hydrogen, by weight, 11 parts, and by measure, 2 parts; but pure water is not attainable, nor is it to be found in the laboratory of the chemist. Fortunately, however, pure water is not necessary, nor even desirable, for either household or manufacturing purposes; because the presence of air and other gases adds very materially to the ease with which steam may be generated, while the ammonia, which most water contains, improves it for manufacturing purposes.

The specific gravity of all waters is not the same. Sea water varies from 1·0269 to 1·0285, the mean being 1·0277, thus requiring 34·9741 cubic feet of sea water to make one ton, and about 35 feet of fresh water. Water is heavier at night than during the day, owing to the atmosphere being more dense, and the additional weight of the dew.

Water has the greatest specific heat of all known liquids except hydrogen, and is therefore taken as the standard for all solids and fluids. The latent heat of water is 143° Fah., and that of ice 140°, as it absorbs that amount of heat in changing from a liquid to a solid state.

Water, under the influence of heat, can be changed from the liquid to the gaseous state in two ways only, either by conversion into steam, or by decomposition into its constituent gases, hydrogen

and oxygen, which decomposition can be effected only at the expense of the apparatus in which it is effected.

TABLE

SHOWING THE QUANTITY AND WEIGHT OF WATER IN PIPES ONE FATHOM IN LENGTH (6 FEET), AND OF DIFFERENT DIAMETERS FROM 1 TO 12 INCHES.

Diameter in Inches.	Quantity in Cubic Inches.	Quantity in Imperial Gallons.	Weight in Lbs. Avoirdupois.
½	14·14	0·051	0·51
1	56·55	0·205	2·05
1½	127·23	0·460	4·60
2	226·19	0·818	8·18
2½	353·43	1·278	12·78
3	508·94	1·841	18·41
3½	692·72	2·506	25·06
4	904·78	3·272	32·72
4½	1145·11	4·142	41·42
5	1413·72	5·113	51·13
5½	1710·60	6·187	61·87
6	2035·75	7·363	73·63
6½	2389·18	8·641	86·41
7	2770·88	10·022	100·22
7½	3180·86	11·505	115·05
8	3619·11	13·090	130·90
8½	4085·64	14·777	147·77
9	4580·44	16·567	165·67
9½	5103·52	18·459	184·59
10	5654·87	20·453	204·53
10½	6234·49	22·550	225·50
11	6842·39	24·748	247·48
11½	7478·56	27·049	270·49
12	8143·01	29·452	294·52
12½	8835·74	38·32	319·50
13	9556·74	55·3	345·00
13½	10306·01	59·6	373·50
14	11083·56	65·2	400·50

TABLE

SHOWING THE QUANTITY OF WATER PER LINEAL FOOT IN PUMPS, OR VERTICAL PIPES OF DIFFERENT DIAMETERS.

Diameter of Pump in Inches.	Number of Gallons per Lineal Foot.	Number of Cubic Feet per Lineal Foot.	Diameter of Pump in Inches.	Number of Gallons per Lineal Foot.	Number of Cubic Feet per Lineal Foot.
2	·136	·0218	8	2·176	·3490
2¼	·172	·0276	8¼	2·314	·3712
2½	·212	·0340	8½	2·456	·3940
2¾	·257	·0412	8¾	2·603	·4175
3	·306	·0490	9	2·754	·4417
3¼	·359	·0576	9¼	2·909	·4666
3½	·416	·0668	9½	3·068	·4923
3¾	·478	·0766	9¾	3·232	·5184
4	·544	·0872	10	3·400	·5454
4¼	·614	·0985	10¼	3·572	·5730
4½	·688	·1104	10½	3·748	·6013
4¾	·767	·1230	10¾	3·929	·6302
5	·850	·1363	11	4·114	·6599
5¼	·937	·1503	11¼	4·303	·6902
5½	1·028	·1649	11½	4·496	·7212
5¾	1·124	·1803	11¾	4·694	·7529
6	1·224	·1963	12	4·896	·7853
6¼	1·328	·2130	12½	5·312	·8521
6½	1·436	·2304	13	5·746	·9217
6¾	1·549	·2489	13½	6·196	·9939
7	1·666	·2672	14	6·664	1·0689
7¼	1·787	·2866	15	7·650	1·2271
7½	1·912	·3067	16	8·704	1·3962
7¾	2·042	·3275	18	11·016	1·7670

One cubic foot of water weighs 62½ lbs., and contains 7½ U. S. gallons.

One cubic foot of ice weighs 57 lbs.

TABLE

SHOWING THE WEIGHT OF WATER AT DIFFERENT TEMPERATURES.

Temperature, Fah.	Weight of a Cubic Foot in Lbs.	Temperature, Fah.	Weight of a Cubic Foot in Lbs.
40°	62·408	172°	60·72
42°	62·406	182°	60·55
52°	62·377	192°	60·28
62°	62·321	202°	60·05
72°	62·025	212°	59·82
82°	62·015	230°	59·37
92°	62·004	250°	58·85
102°	61·092	275°	58·17
112°	61·078	300°	57·42
122°	61·063	350°	55·94
132°	61·047	400°	54·34
142°	61·030	450°	52·70
152°	61·011	500°	51·02
162°	60·092	600°	47·64

TABLE

SHOWING THE BOILING POINT FOR FRESH WATER AT DIFFERENT ALTITUDES ABOVE SEA-LEVEL.

Boiling Point in Deg. Fah.	Altitude above Sea-Level in Feet.	Boiling Point in Deg. Fah.	Altitude above Sea-Level in Feet	Boiling Point in Deg. Fah.	Altitude above Sea-Level in Feet.
184°	15,221	195°	9,031	206°	3,115
185	14,649	196	8,481	207	2,589
186	14,075	197	7,932	208	2,063
187	13,498	198	7,381	209	1,539
188	12,934	199	6,843	210	1,025
189	12,367	200	6,304	211	512
190	11,799	201	5.764	212	Sea-Level = 0
191	11,243	202	5,225		
192	10,685	203	4,697	Below Sea-Level	
193	10,127	204	4,169	213	511
194	9,579	205	3,642		

TABLE

SHOWING THE CAPACITY OF CISTERNS AND TANKS. COMPUTED IN BARRELS OF $31\frac{1}{2}$ GALLONS.

Depth.	Diameter in Feet.															
	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
5	23·3	33·6	45·7	59.7	75·5	93·2	112·8	134·3	157·6	182·8	209·8	238·7	269·5	302·1	336·6	373·0
6	28·0	40·3	54·8	71·7	90·6	111·9	135·4	161·1	189·1	219·3	251·8	286·5	323·4	362·6	404·0	447·6
7	32·7	47·0	64·0	83·6	105·7	130·6	158·0	188·0	220·6	255·9	293·7	334·2	377·3	423·0	471·3	522·2
8	37·3	53·7	73·1	95·5	120·9	149·2	180·5	214·8	252·1	292·4	335·7	382·0	431·2	483·4	538·6	596·8
9	42·0	60·4	82·2	107·4	136·0	167·9	203·1	241·7	283·7	329·0	377·7	429·7	485·1	543·8	605·9	671·4
10	46·7	67·1	91·4	119·4	151·1	186·5	225·7	268·6	315·2	365·5	419·6	477·4	539·0	604·3	673·3	746·0
11	51·3	73·9	100·5	131·3	166·2	205·1	248·2	295·4	346·7	402·1	461·6	525·2	592·9	664·7	740·6	820·6
12	56·0	80·6	109·7	143·2	181·3	223·8	270·8	322·3	378.2	438·6	503·5	572·9	646·8	725·1	807·9	895·2
13	60·7	87·3	118·8	155·2	196.4	242·4	293·4	349·1	409·7	475·2	545·5	620·7	700·7	785·5	875·2	969·8
14	65·3	94·0	127·9	167·1	211·5	261·1	315·9	376·0	441·3	511·8	587·5	668·4	754·6	846·0	942·6	1044·4
15	70·0	100·7	137·1	179·0	226·6	279·8	338·5	402·8	472·8	548·3	629·4	716·2	808·5	906·4	1009·9	1119·0
16	74·7	107·4	146·2	191·0	241·7	298·4	361·1	429·7	504·3	584·9	671·4	763·9	862·4	966·8	1077·2	1193·6
17	79·3	114·1	155·4	202·9	256·8	317·0	383·6	456·6	535·8	621·4	713·4	811·6	916·3	1027·2	1144·6	1268·2
18	84·0	120·9	164·5	214·8	272·0	335·7	406·2	483·4	567·3	658·0	755·3	859·4	970·2	1087·7	1211·9	1342·8
19	88·7	127·6	173·6	226·8	287·0	354·3	428·8	510·3	598·9	694·5	797·3	907·1	1024·1	1148·1	1279·2	1417·4
20	93·3	134·3	182·8	238·7	302·1	373·0	451·3	537·1	630·4	731·1	839·3	954·9	1078·0	1208·5	1346·5	1492·0

TABLE

SHOWING THE POWER REQUIRED TO RAISE WATER TO DIFFERENT ALTITUDES, VARYING FROM 1 FOOT TO 20,000 FEET.

What a Horse-Power will raise per Minute.		2 H. P. will raise per Minute.		3 H. P. will raise per Minute.		4 H. P. will raise per Minute.		5 H. P. will raise per Minute.		6 H. P. will raise per Minute.		7 H. P. will raise per Minute.		8 H. P. will raise per Minute.	
Gals. or Feet.	Feet or Gals.	Gals. or Feet.	Feet or Gals.	Gals. or Feet.	Feet or Gals.	Gals. or Feet.	Feet or Gals.	Gals. or Feet.	Feet or Gals.	Gals. or Feet.	Feet or Gals.	Gals. or Feet.	Feet or Gals.	Gals. or Feet.	Feet or Gals.
2,500	1	5,000	1	7,500	1	10,000	1	12,500	1	15,000	1	17,500	1	20,000	1
1,250	2	2,500	2	3,750	2	5,000	2	6,250	2	7,500	2	8,750	2	10,000	2
833	3	1,666	3	2,500	3	3,333	3	4,166	3	5,000	3	5,833	3	6,666	3
625	4	1,250	4	1,875	4	2,500	4	3,125	4	3,750	4	4,375	4	5,000	4
500	5	1,000	5	1,500	5	2,000	5	2,500	5	3,000	5	3,500	5	4,000	5
416	6	833	6	1,250	6	1,666	6	2,083	6	2,500	6	2,916	6	3,333	6
357	7	714	7	1,070	7	1,428	7	1,785	7	2,142	7	2,500	7	2,857	7
312	8	625	8	937	8	1,250	8	1,562	8	1,875	8	2,187	8	2,500	8
277	9	555	9	833	9	1,111	9	1,388	9	1,666	9	1,944	9	2,222	9
250	10	500	10	750	10	1,000	10	1,250	10	1,500	10	1,750	10	2,000	10
125	20	250	20	375	20	500	20	625	20	750	20	875	20	1,000	20
83	30	166	30	250	30	333	30	416	30	500	30	583	30	666	30
62	40	125	40	187	40	250	40	312	40	375	40	437	40	500	40
50	50	100	50	150	50	200	50	250	50	300	50	350	50	400	50
41	60	83	60	125	60	166	60	208	60	250	60	291	60	333	60
35	70	71	70	107	70	142	70	178	70	214	70	250	70	285	70
31	80	62	80	93	80	125	80	156	80	187	80	218	80	250	80
27	90	55	90	83	90	111	90	138	90	166	90	194	90	222	90
25	100	50	100	75	100	100	100	125	100	150	100	175	100	200	100

TABLE

SHOWING THE CAPACITY OF CISTERNS IN GALLONS FOR EACH 10-INCH DEPTH.

Diameter in Feet.	Gallons.	Diameter in Feet.	Gallons.	Diameter in Feet.	Gallons.
2·	19·5	6·5	206·8	12	705·
2·5	30·5	7·	239·8	13	827·4
3·	44·6	7·5	275·4	14	959·7
3·5	59·97	8·	313·3	15	1101·6
4·	78·33	8·5	353·7	20	1958·4
4·5	99·14	9·	396·5	25	3059·9
5·	122·4	9·5	461·4	30	4406·4
5·5	148·1	10·	489·6	35	5990·
6·	176·2	11·	592·4	40	7831·

Rule *for finding the horse-power of waterfalls.*—Multiply the area of the cross-section of the waterfall in feet by its velocity in feet per minute; this product will give the number of cubic feet flowing through per minute. Multiply this by $62\frac{1}{2}$ lbs., the number of pounds in a cubic foot of water. Multiply this last product by the fall in feet, and divide by 33,000. The quotient will be the horse-power of the waterfall.

Example.—With a stream or flume 10 feet; depth, 4 feet; area of cross-section, 40 feet; velocity in feet per minute, 150. Then $40 \times 150 = 6{,}000$ cubic feet of water per minute; $6{,}000 \times 62\frac{1}{2} = 375{,}000$ pounds of water per minute. $10 \times 375{,}000 = 3{,}750{,}000$ foot-pounds of the waterfall; $3{,}750{,}000 \div 33{,}000 = 113\frac{7}{11}$ horse-power of the waterfall.

Rule *for finding the contents of an elliptic or oval tank in cubic feet or gallons.*—Multiply the long diameter in inches by the short diameter in inches, this product by ·7854, and this last product by the height of the tank in inches; then divide by 1728, and the result will be the contents of the tank in cubic feet, which, if multiplied by 7·5, gives the number of U. S. gallons in the tank.

Rule *for finding the quantity of water which any square or rectangular box, or tank, is capable of containing in cubic feet or U. S. gallons.*— Multiply the length of the sides in inches by their height in inches; then multiply the width of the ends in inches by their height in inches. Add these two products together and divide by 1728; the product will be the contents in cubic feet. This result being multiplied by 7·5 gives the cubical contents in U. S. gallons.

Rule *for finding the cubical contents of a triangular tank.*— Multiply the length of the base by half its height; multiply this by .7854, then divide the product by 1728; the quotient will be the number of cubic feet, which, if multiplied by 7·5, will give the number of U. S. gallons in the tank.

TABLE

SHOWING THE DAILY AVERAGE NUMBER OF GALLONS OF WATER PER INDIVIDUAL IN DIFFERENT CITIES INCLUDING THE QUANTITY USED FOR MANUFACTURING PURPOSES, FOUNTAINS, ETC.

Washington, D. C.	158	Toronto	77
New York	100	London, England.	·29
Brooklyn.	50	Liverpool, "	23
Philadelphia	55	Glasgow, Scotland	50
Baltimore	40	Edinburgh, "	38
Chicago	75	Dublin, Ireland	25
Boston	60	Paris, France	28
Albany, N. Y.	80	Tours, "	22
Detroit	83	Toulouse, "	26
Jersey City, N. J.	99	Lyons, "	20
Buffalo, N. Y.	61	Leghorn, Italy.	30
Cleveland	40	Berlin, Prussia	20
Columbus	30	Hamburg, "	33
Montreal	55	Richmond, Va.	36
San Francisco	42	City of Mexico	25
Hartford, Conn.	32	Vienna	20
Charleston, S. C.	27	St. Petersburg	19

Vapors.

The mechanical properties of vapor are similar to those of gases in general. When a vapor or gas is contained in a close vessel, the inner surface of the vessel will sustain a pressure arising from the elasticity of the fluid. This pressure is produced by the mutual repulsion of the particles, which gives them a tendency to fly asunder, and causes the mass of the fluid to exert a force tending to burst any vessel within which it is confined. This pressure is uniformly diffused over every part of the surface of the vessel in which such fluid is contained. It is to this quality that all the mechanical power of steam is due.

It is well known from common observation, that liquids, if not confined in close vessels, become transformed into a condition resembling gases at ordinary temperatures, and then disappear. This transformation takes place in nearly all liquids more or less rapidly at ordinary temperatures, though in some it takes place at a very low temperature and at an imperceptible rate of evaporation; the lighter the specific gravity, the more rapidly it will disappear.

TABLE

SHOWING THE TEMPERATURE OF SATURATED VAPOR IN ATMOSPHERES, ACCORDING TO ZEUNER.

Atmospheres.	Temperature in Degrees Fah., Water.	Atmospheres.	Temperature in Degrees Fah., Water.
1	212°	6	318·5°
2	248°	7	329·5°
3	272°	8	339·0°
4	291°	9	348·0°
5	306°	10	357·0°

TABLE

SHOWING THE PRESSURE AND TEMPERATURE OF THE VAPORS OF WATER FROM 32° TO 400° FAH.

Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.	Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.
32	0·006	0·09	0·00	73	0·026	0·38	0·02
33	0·006	0·09	0·00	74	0·027	0·40	0·01
34	0·006	0·09	0·01	75	0·028	0·41	0·02
35	0·007	0·10	0·00	76	0·029	0·43	0·03
36	0·007	0·10	0·01	77	0·031	0·46	0·00
37	0·007	0·11	0·01	78	0·031	0·46	0·01
38	0·008	0·12	0·00	79	0·032	0·47	0·00
39	0·008	0·12	0·01	80	0·032	0·47	0·02
40	0·009	0·13	0·00	81	0·033	0·49	0·01
41	0·009	0·13	0·01	82	0·034	0·50	0·03
42	0·009	0·14	0·00	83	0·036	0·53	0·01
43	0·009	0·14	0·01	84	0·037	0·54	0·03
44	0·010	0·15	0·00	85	0·039	0·57	0·03
45	0·010	0·15	0·01	86	0·041	0·60	0·02
46	0·011	0·16	0·00	87	0·042	0·62	0·01
47	0·011	0·16	0·01	88	0·043	0·63	0·02
48	0·011	0·17	0·00	89	0·044	0·65	0·03
49	0·011	0·17	0·01	90	0·046	0·68	0·03
50	0·012	0·18	0·00	91	0·048	0·71	0·01
51	0·012	0·18	0·01	92	0·049	0·72	0·02
52	0·013	0·19	0·01	93	0·050	0·74	0·02
53	0·013	0·20	0·01	94	0·052	0·76	0·05
54	0·014	0·21	0·00	95	0·055	0·81	0·03
55	0·014	0·21	0·01	96	0·057	0·84	0·01
56	0·015	0·22	0·02	97	0·058	0·85	0·05
57	0·016	0·24	0·00	98	0·061	0·90	0·00
58	0·016	0·24	0·01	99	0·061	0·90	0·03
59	0·017	0·25	0·01	100	0·063	0·93	0·04
60	0·018	0·26	0·00	101	0·066	0·97	0·01
61	0·018	0·26	0·00	102	0·067	0·98	0·03
62	0·018	0·26	0·02	103	0·069	1·01	0·05
63	0·019	0·28	0·01	104	0·072	1·06	0·03
64	0·020	0·29	0·00	105	0·074	1·09	0·06
65	0·020	0·29	0·02	106	0·078	1·15	0·01
66	0·021	0·31	0·01	107	0·079	1·16	0·02
67	0·022	0·32	0·02	108	0·080	1·18	0·04
68	0·023	0·34	0·00	119	0·083	1·22	0·04
69	0·023	0·34	0·01	110	0·086	1·26	0·03
70	0·024	0·35	0·00	111	0·088	1·29	0·05
71	0·024	0·35	0·02	112	0·091	1·34	0·03
72	0·025	0·37	0·01	113	0·093	1·37	0·03

TABLE — (*Continued.*)

Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.	Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.
114	0·095	1·40	0·07	157	0·300	4·41	0·09
115	0·100	1·47	0·03	158	0·306	4·50	0·03
116	0·102	1·50	0·04	159	0·308	4·53	0·14
117	0·105	1·54	0·03	160	0·318	4·67	0·11
118	0·107	1·57	0·08	161	0·325	4·78	0·19
119	0·112	1·65	0·03	162	0·338	4·97	0·10
120	0·114	1·68	0·04	163	0·345	5·07	0·12
121	0·117	1·72	0·06	164	0·353	5·19	0·12
122	0·121	1·78	0·04	165	0·361	5·31	0·14
123	0·124	1·82	0·05	166	0·371	5·45	0·12
124	0·127	1·87	0·04	167	0·379	5·57	0·02
125	0·130	1·91	0·09	168	0·387	5·69	0·13
126	0·136	2·00	0·04	169	0·396	5·82	0·13
127	0·139	2·04	0·05	170	0·405	5·95	0·15
128	0·142	2·09	0·06	171	0·415	6·10	0·15
129	0·146	2·15	0·05	172	0·425	6·25	0·16
130	0·150	2·20	0·06	173	0·436	6·41	0·13
131	0·154	2·26	0·06	174	0·445	6·54	0·15
132	0·158	2·32	0·06	175	0·455	6·69	0·16
133	0·162	2·38	0·06	176	0·466	6·85	0·15
134	0·166	2·44	0·09	177	0·476	7·00	0·15
135	0·172	2·53	0·04	178	0·488	7·15	0·21
136	0·175	2·57	0·09	179	0·501	7·36	0·18
137	0·181	2·66	0·03	180	0·513	7·54	0·12
138	0·183	2·69	0·12	181	0·521	7·66	0·15
139	0·191	2·81	0·07	182	0·531	7·81	0·17
140	0·196	2·88	0·06	183	0·543	7·98	0·19
141	0·200	2·94	0·07	184	0·556	8·17	0·19
142	0·205	3·01	0·08	185	0·569	8·36	0·18
143	0·210	3·09	0·10	186	0·581	8·54	0·18
144	0·217	3·19	0·04	187	0·593	8·72	0·17
145	0·220	3·23	0·09	188	0·605	8·89	0·19
146	0·226	3·32	0·08	189	0·618	9·08	0·21
147	0·231	3·40	0·10	190	0·631	9·29	0·21
148	0·238	3·50	0·10	191	0·646	9·50	0·22
149	0·245	3·60	0·09	192	0·661	9·72	0·22
150	0·251	3·69	0·02	193	0·676	9·94	0·22
151	0·259	3·81	0·09	194	0·691	10·16	0·20
152	0·265	3·90	0·08	195	0·705	10·36	0·21
153	0·271	3·98	0·11	196	0·719	10·57	0·22
154	0·278	4·09	0·10	197	0·734	10·79	0·22
155	0·285	4·19	0·06	198	0·749	11·01	0·22
156	0·289	4·25	0·16	199	0·764	11·23	0·22

TABLE — (*Continued.*)

Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.	Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.
200	0·770	11·45	0·12	243	1·794	26·37	0·47
201	0·787	11·57	0·25	244	1·826	26·84	0·49
202	0·804	11·82	0·43	245	1·859	27·33	0·50
203	0·833	12·25	0·25	246	1·893	27·83	0·51
204	0·850	12·50	0·24	247	1·928	28·34	0·50
205	0·867	12·74	0·27	248	1·962	28·84	0·52
206	0·885	13·01	0·28	249	1·997	29·36	0·51
207	0·904	13·29	0·28	250	2·032	29·87	0·53
208	0·923	13·57	0·28	251	2·068	30·40	0·53
209	0·942	13·85	0·28	252	2·104	30·93	0·54
210	0·961	14·13	0·29	253	2·141	31·47	0·56
211	0·981	14·42	0·28	254	2·179	32·03	0·57
212	1·000	14·77	0·30	255	2·217	32·60	0·56
213	1·020	15·00	0·29	256	2·256	33·16	0·56
214	1·040	15·29	0·31	257	2·294	33·72	0·59
215	1·061	15·60	0·32	258	2·334	34·31	0·59
216	1·083	15·92	0·31	259	2·374	34·90	0·60
217	1·104	16·23	0·31	260	2·415	35·50	0·60
218	1·125	16·54	0·32	261	2·456	36·10	0·62
219	1·147	16·86	0·32	262	2·498	36·72	0·63
220	1·169	17·18	0·33	263	2·541	37·35	0·63
221	1·191	17·51	0·35	264	2·584	37·98	0·64
222	1·215	17·86	0·34	265	2·627	38·62	0·64
223	1·238	18·20	0·34	266	2·671	39·26	0·67
224	1·261	18·54	0·35	267	2·716	39·93	0·66
225	1·285	18·89	0·35	268	2·761	40·59	0·67
226	1·309	19·24	0·37	269	2·807	41·26	0·69
227	1·334	19·61	0·37	270	2·854	41·95	0·69
228	1·359	19·98	0·38	271	2·901	42·64	0·71
229	1·385	20·36	0·44	272	2·949	43·35	0·71
230	1·415	20·80	0·40	273	2·997	44·06	0·72
231	1·442	21·20	0·39	274	3·046	44·78	0·75
232	1·469	21·59	0·42	275	3·097	45·53	0·73
233	1·497	22·01	0·41	276	3·147	46·26	0·75
234	1·525	22·42	0·41	277	3·198	47·01	0·77
235	1·553	22·83	0·43	278	3·250	47·78	0·77
236	1·582	23·26	0·44	279	3·303	48·55	0·80
237	1·612	23·70	0·42	280	3·357	49·35	0·79
238	1·641	24·12	0·43	281	3·411	50·14	0·81
239	1·670	24·55	0·44	282	3·466	50·95	0·81
240	1·700	24·99	0·45	283	3·521	51·76	0·81
241	1·731	25·45	0·45	284	3·576	52·57	0·82
242	1·762	25·90	0·47	285	3·632	53·39	0·84

TABLE — (*Continued.*)

Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.	Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.
286	3·689	54·23	0·85	329	6·945	102·09	1·44
287	3·747	55·08	0·87	330	7·043	103·53	1·44
288	3·806	55·95	0·88	331	7·141	104·97	1·44
289	3·866	56·83	0·88	332	7·239	106·41	1·46
290	3·926	57·71	0·90	333	7·338	107·87	1·45
291	3·987	58·61	0·91	334	7·437	109·32	1·46
292	4·049	59·52	0·93	335	7·536	110·78	1·50
293	4·112	60·45	0·92	336	7·638	112·28	1·51
294	4·175	61·37	0·94	337	7·741	113·79	1·55
295	4·239	62·31	0·96	338	7·846	115·34	1·55
296	4·304	63·27	0·97	339	7·952	116·89	1·59
297	4·370	64·24	0·98	340	8·060	118·48	1·60
298	4·437	65·22	1·00	341	8·169	120·08	1·62
299	4·505	66·22	1·02	342	8·279	121·70	1·62
300	4·574	67·24	1·01	343	8·389	123·32	1·63
301	4·643	68·25	1·02	344	8·500	124·95	1·65
302	4·712	69·27	1·03	345	8·612	126·60	1·64
303	4·782	70·30	1·04	346	8·724	128·24	1·68
304	4·853	71·34	1·06	347	8·838	129·92	1·69
305	4·925	72·40	1·07	348	8·953	131·61	1·72
306	4·998	73·47	1·09	349	9·070	133·33	1·75
307	5·072	74·56	1·10	350	9·189	135·08	1·78
308	5·147	75·66	1·12	351	9·310	136·86	1·81
309	5·223	76·78	1·13	352	9·433	138·67	1·80
310	5·300	77·91	1·16	353	9·556	140·47	1·83
311	5·379	79·07	1·16	354	9·680	142·30	1·82
312	5·458	80·23	1·18	355	9·804	144·12	1·84
313	5·538	81·41	1·19	356	9·929	145·96	1·85
314	5·619	82·60	1·20	357	10·055	147·81	1·87
315	5·701	83·80	1·22	358	10·182	149·68	1·89
316	5·784	85·02	1·22	359	10·311	151·57	1·93
317	5·867	86·24	1·24	360	10·442	153·50	1·95
318	5·951	87·48	1·23	361	10·575	155·45	1·99
319	6·035	88·71	1·27	362	10·710	157·44	2·01
320	6·121	89·98	1·28	363	10·847	159·45	2·03
321	6·208	91·26	1·29	364	10·985	161·48	2·03
322	6·296	92·55	1·31	365	11·123	163·51	2·04
323	6·385	93·86	1·32	366	11·262	165·55	2·04
324	6·475	95·18	1·34	367	11·401	167·59	2·08
325	6·556	96·42	1·35	368	11·542	169·67	2·08
326	6·658	97·87	1·37	369	11·684	171·75	2·14
327	6·751	99·24	1·40	370	11·829	173·89	2·16
328	6·846	100·64	1·45	371	11·976	176·05	2·19

TABLE — (*Continued.*)

Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.	Temp. Fah.	Pressures in Atmospheres.	Pressures in Lbs. per Sq. Inch.	Differences.
372	12·125	178·24	2·19	387	14·510	213·30	2·51
373	12·274	180·43	2·22	388	14·681	215·81	2·54
374	12·425	182·65	2·22	389	14·854	218·35	2·58
375	12·576	184·87	2·23	390	15·029	220·93	2·60
376	12·728	187·10	2·25	391	15·206	223·53	2·56
377	12·881	189·35	2·26	392	15·380	226·09	2·13
378	13·035	191·61	2·28	393	15·457	227·22	2·64
379	13·190	193·86	2·31	394	15·637	229·86	2·69
380	13·347	196·20	2·35	395	15·820	232·55	2·65
381	13·507	198·55	2·40	396	16·000	235·20	2·79
382	13·670	200·95	2·44	397	16·190	237·99	2·87
383	13·836	203·39	2·45	398	16·385	240·86	2·90
384	14·003	205·84	2·47	399	16·582	243·76	2·94
385	14·171	208·31	2·49	400	16·782	246·70	3·04
386	14·340	210·80	2·50				

Gases.

All substances, whether animal, vegetable, or mineral, consisting of carbon, hydrogen, and oxygen, when exposed to a red heat, produce various inflammable elastic fluids capable of furnishing artificial light. The products of perfect combustion are gases which form in accordance with unchangeable laws. Many of the gases have already been brought into the liquid state by the conjoint agency of cold and compression, and all of them are probably susceptible of a similar reduction by the use of means sufficiently powerful for the required end. They must consequently be regarded as the superheated steams or vapors of the liquids into which they are compressed.

When a gas or vapor is compressed into half its original bulk, its pressure is double; when compressed into a third of its original bulk, its pressure is trebled; when compressed into a fourth of its original bulk, its pressure is quadrupled; and generally the pressure varies inversely as the bulk into which the gas is compressed. So in like manner, if the volume be doubled, the pressure is made

one-half of what it was before,— the pressure in every case being reckoned from 0, or from a perfect vacuum.

Thus, if we take the average pressure of the atmosphere at 14·7 pounds on the square inch, a cubic foot of air, if suffered to expand into twice its bulk, by being placed in a vacuum measuring two cubic feet, will have a pressure of 7·35 pounds above a perfect vacuum, and also of 7·35 pounds below the atmospheric pressure; whereas, if the cubic foot be compressed into a space of half a cubic foot, the pressure will become 29·4 pounds above a perfect vacuum, and 14·7 above the atmospheric pressure. The specific gravity of any one gas to that of another will not exactly conform to the same ratio under different degrees of heat, and other pressures of the atmosphere.

Oxygen is the name given to the solid particles of oxygen gas, which is a combination of oxygen, caloric, and light, and is the simplest form in which oxygen can be obtained. Oxygen is called the radical or base of the gas; and the same mode of expression is used in other cases. Oxygen enters into chemical combination with a great number of substances, in which it exists in a concrete or solid state; it is by the application of heat, or of acids, to some of the substances containing it, that it is usually procured in the form of gas. Oxygen gas is the only one that can be breathed by animals for any length of time with impunity. The power of atmospheric air in supporting respiration is owing to the oxygen. Oxygen combines with all the metals, and in this state they are called metallic oxides, depriving them of their metallic lustre, and giving them an earthy or rusty appearance. Any of the metals are capable of combining with different proportions of oxygen. Those with one proportion are called *protoxides*; of two, *deutoxides*; those of three, *tritoxides*.

Nitrogen.—Nitrogen gas is most easily described by including many of its negative qualities. It has no taste; it unites with oxygen in several proportions; it also unites with hydrogen. Though incapable of being breathed above its base, nitrogen is a component portion of all animal substances; it is lighter than oxy-

gen. Nitrogen gas may be variously obtained. If the oxygen be extracted from the atmospheric air, this substance will remain, and will generally be very pure, unless the oxygen has been extracted by respiration. If iron filings and sulphur, moistened with water, be put into a jar containing atmospheric air, this gas will in a day or two be all the air that remains in the jar, as the oxygen will be absorbed by the iron and sulphur. Phosphorus or sulphuret of lime or potass, inclosed with common air in a jar, will produce a similar effect.

Hydrogen.—Hydrogen, like oxygen and nitrogen, is invisible, elastic, and inodorous; but the last quality it seldom possesses, because it is very seldom perfectly dry, and when it contains water in solution, like alkaline sulphurets, its odor is considerably fetid. Hydrogen with oxygen forms water; and it is by the decomposition of water that chemists obtain it in the greatest abundance and purity. For this purpose iron filing or turnings, or granulated zinc, are put into a retort, and covered with sulphuric acid diluted with four times its weight in water. A violent effervescence ensues, a large quantity of gas is evolved, and issuing from the retort is collected in the usual manner by the pneumatic apparatus. In this experiment the acid is not decomposed; it is the oxygen of the water with which the acid is diluted that seizes upon and oxidizes the metal, and the hydrogen, in the same portion of water being thus disengaged, passes over in the state of gas. The hydrogen obtained by using zinc is the purest, that obtained by using iron generally containing some carbon.

Hydrogen combines with a larger quantity of oxygen than any other body; its combustion, therefore, when mixed with oxygen, produces a more intense heat than any other combustion.

Carbon.—Vegetables, when burnt or distilled in close vessels till their volatile parts are entirely separated, leave a black, brittle and cinereous substance which constitutes the greater part of the woody fibre, and is called *charcoal.* Charcoal contains a portion of earthy and saline impurities, but, when entirely freed from these and other impurities, a solid, simple, combustible substance re-

mains, which is called carbon. Carbon exists naturally in a state of greater purity than can be prepared by art. The diamond is pure carbon crystallized, and when pure is colorless and transparent. It is the hardest substance known; and, as it sustains a considerable degree of heat unchanged, it was formerly considered to be incombustible. It may, however, be consumed by a burning-glass, and even by the heat of a furnace. The difficulty of burning it appears to arise from its hardness; for Morveau and Tennant have rendered common charcoal so hard, by exposing it for some time to a violent fire in close vessels, that it endured a red heat without catching fire. Common charcoal contains only 64 parts of diamond, or pure carbon, and 36 of oxygen in every 100.

The common charcoal of commerce is usually prepared from young wood, which is piled up near the place where it is cut in conical heaps, covered with earth, and burnt with the least possible access of air. When the fire is supposed to have penetrated to the centre of the thickest pieces, it is extinguished by entirely closing the vents. When charcoal is wanted very pure, the product of this mode of preparing it will not suffice; for the manufacturing of the best gunpowder, it is distilled in iron cylinders. Chemists prepare it in small quantities in a crucible covered with sand, and after they have thus prepared it, they pound it, and wash away the salts it contains by muriatic acid; the acid is removed by the plentiful use of water, and afterwards the charcoal is exposed to a low red heat. Pure charcoal is perfectly tasteless and insoluble in water.

Charcoal newly prepared absorbs moisture with avidity. It also absorbs oxygen and other gases, which are condensed in its pores in quantity many times exceeding its own bulk, and which are given out unaltered. Fresh charcoal allowed to cool without exposure to air, and the gas then admitted, will absorb 2·25 times its bulk of atmospheric air immediately, and 75 per cent. more in four or five hours; of oxygen gas about 1·8 immediately, and slowly one more; of nitrogen gas, 1·65 immediately.

Technical and Chemical Terms as Applied to Substances that bear Relations to the Steam-Engine both in Theory and Practice.

Alkali, or antacid, means any substance which, when mingled with acid, produces fermentation.

Ammonia. — This alkali, when perfectly caustic, enables chemists to distinguish between the salts of lime and those of magnesia, as it precipitates the earth from the latter salts, but not from the former.

Analysis means resolution, by chemistry, of any matter into its primary and constituent parts.

Atoms. — In the chemical combination of bodies with each other, it is observed that some unite in all proper proportions; others in all proportions as far as a certain point beyond which combination no longer takes place. There are also many examples in which they unite in one proportion only, and others in several proportions; and these proportions are definite, and in the intermediate ones no combination ensues.

Bases. — This term is usually applied to alkalies, earths, and metallic oxides in their relations to the acids and salts. It is sometimes also applied to the particular constituents of an acid or oxide, on the supposition that the substance combined with the oxygen, etc., is the basis of the compound to which it owes its particular qualities.

Calcination. — This term is applied to the fixed residues of such matters as have undergone combustion, and are called cinders in common language, and oxides by chemists. This operation, when considered with regard to these residues, is termed calcination.

Combination is understood to be the intimate union of the par-

ticles of different substances by chemical attraction, so as to form a compound possessed of new and peculiar properties.

Compound. — A compound is the result or effect of a composition of different things, or that which arises from them. It stands opposed to simple.

Equivalents are terms introduced into chemistry to express the system of definite ratios in which the molecular atoms of this science reciprocally unite.

Evaporation is a chemical process usually performed by applying heat to any compound substance, in order to dispel the volatile parts.

Fixed. — This epithet is applied to such bodies as so far resist the action of heat so as not to rise in vapor. It is the opposite of volatile; but it must be observed that the fixity of bodies is merely a relative term, as an adequate degree of heat will dissipate all.

Neutral. — A term applied to saline compounds of an acid or alkali nature, which are so called, because they do not possess the characters of acid or alkaline salts.

Neutralization. — This term is applied when acid and alkaline matter are combined in such proportion that the compound does not change the color of litmus or violets, in which condition they are said to be neutralized.

Oxide. — Any substance which combines with oxygen without being in the state of an acid is an oxide.

Oxidation. — This term is applied to the process of converting metals and other substances into oxides by combining with them a certain portion of oxygen.

Phosphate is a salt formed by the union of phosphoric acid

with salifiable bases; thus, phosphate of ammonia, phosphate of lime, etc.

Pyrites. — Substances which strike fire when rubbed or thrown together. They are frequently found in bituminous coal, and often induce spontaneous combustion.

Saline. — A term applied to any substance of a salty nature. The number of saline substances is very considerable, and they possess peculiar characters by which they are distinguished from other substances.

Saturation. — A term applied to bodies which have a chemical affinity for each other, and which will only unite in certain proportions. When, therefore, a fluid has dissolved as much of any substance as it is capable of dissolving, it is said to have reached the point of saturation. Thus, water will dissolve one-quarter of its weight of common salt, and if more salt be added, it will sink to the bottom in a solid state.

Areas of Circles.

The term area means any opening or flat surface confined between any lines; a definite space; superficial contents of any figure; any plain space or surface included within any given lines; but when used in connection with the steam-engine, it means the number of square inches in the piston, or valve, against which the steam acts, as the case may be.

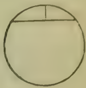
A circle may be considered as composed of many triangles, whose bases are the circumference of the circle, and whose vertices are coincident with the centre of the circle. If a cylinder be drawn, whose height equals $\frac{1}{4}$ its diameter, the convex surface of such a cylinder is just equal to the area of the circle. A circular vessel will contain a greater quantity than a vessel of any other shape made of the same amount of material. The areas of circles are to each other as the square of their diameters. The diameter of a circle being 1, its circumference equals 3·1416.

The diameter of a circle is a straight line drawn through its centre, touching both sides, thus..............

The radius of a circle is half the diameter.............

A chord is a straight line joining any two places in the circumference of a circle..........................

The versed sine is a perpendicular joining the middle of the chord and circumference of a circle...............



An arc is any part of the circumference of a circle...

A triangle has 3 sides and 3 angles.......................

A parallelogram has 4 sides and 4 angles..................

A pentagon has 5 sides and 5 angles.....................

A hexagon has 6 sides and 6 angles......................

A heptagon has 7 sides and 7 angles......................

An octagon has 8 sides and 8 angles......................

A nonagon has 9 sides and 9 angles........................ 

A decagon has 10 sides and 10 angles........................ 

An endecagon has 11 sides and 11 angles.................... 

A dodecagon has 12 sides and 12 angles..................... 

Rules.

To find the circumference of a circle, multiply the diameter by 3 1416; the product is the circumference.

To find the diameter of a circle, divide the circumference by 3·1416, the quotient is the diameter; or multiply the square root of the area by 1·12837, the product is the diameter.

To find the area of a circle, multiply the square of the diameter by ·7854, the product is the area; or multiply half the circumference by half the diameter, the product is the area; or multiply the diameter by the circumference, and divide by 4; the quotient is the area.

To find the area of an ellipse or oval, multiply the long diameter by the short diameter; multiply this product by ·7854, and the product will be the superficial area of the ellipse.

To find the circumference of an ellipse or oval, multiply ½ the sum of the two diameters by 3·1416; the product will be the circumference of the ellipse.

To find the area of a parallelogram, multiply the length by the height or perpendicular breadth.

To find the area of a triangle, multiply the base by the perpendicular height, and take half the product.

To find the area of a trapezoid, multiply half the sum of the parallel sides by the perpendicular distance between them; the product will be the area.

To find the area of a quadrilateral inscribed in a circle.—From half the sum of the four sides subtract each side severally; multiply the four remainders together; the square root of the product is the area.

To find the area of any quadrilateral figure, divide the quadrilateral into two triangles; the sum of the areas of the triangles is the area.

To find the area of any polygon, divide the polygon into triangles and trapezoids by drawing diagonals; find the areas of these, as above shown, for the area.

To find the area of a regular polygon, multiply half the perimeter of the polygon by the perpendicular drawn from the centre to the centre of one of the sides.

To find the area of a sector of a circle, multiply half the length of the arc of the sector by the radius. Or, multiply the number of degrees in the arc by the square of the radius, and by ·008727.

To find the area of a segment of a circle, find the area of the sector which has the same arc as the segment; also the area of the triangle formed by the radial sides of the sector and the chord of the arc; the difference or the sum of these areas will be the area of the segment, according as it is less or greater than a semicircle.

To find the area of a cycloid, multiply the area of the generating circle by 3.

To find the length of an arc of a parabola cut off by a double ordinate to the axis. — To the square of the ordinate add four-fifths of the square of the absciss; twice the square root of the sum is the length nearly.

To find the area of an ellipse, multiply the product of the two axes by ·7854.

To find the area of an elliptic segment, the base of which is parallel to either axis of the ellipse. — Divide the height of the

segment by the axis of which it is a part, and find the area of a circular segment, which the height is equal to in this quotient; multiply the area thus found by the two axes of the ellipse successively; the product is the area.

To find the length of an arc of a hyperbola beginning at the vertex. — To nineteen times the transverse axis, add twenty-one times the parameter to this axis, and multiply the sum by the quotient of the absciss divided by the transverse.

To find the area of a hyperbola, to the product of the transverse and absciss add five-sevenths of the square of the absciss, and multiply the square root of the sum by twenty-one; to this product add four times the square root of the product of the transverse and absciss; multiply the sum by four times the product of the conjugate and absciss, and divide by seventy-five times the transverse. The quotient is the area nearly.

To find the surface of a prism or a cylinder, the perimeter of the end multiplied by the height gives the upright surface; add twice the area of an end.

To find the cubic contents of a prism or a cylinder, multiply the area of the base by the height.

To find the surface of a pyramid or a cone, multiply the perimeter of the base by half the slant height, and add the area of the base.

To find the cubic contents of a pyramid or a cone, multiply the area of the base by one-third of the perpendicular height.

To find the surface of a frustum of a pyramid or a cone, multiply the sum of the perimeters of the ends by half the slant height, and add the areas of the ends.

To find the cubic contents of a frustum of a pyramid or a cone, add together the areas of the two ends, and the mean proportional between them (that is, the square root of their product), and multiply the sum by one-third of the perpendicular height.

To find the cubic contents of a segment of a sphere, from three times the diameter of the sphere subtract twice the height of the

segment; multiply the difference by the square of the height, and by ·5236.

To find the cubic contents of a frustum or zone of a sphere.—To the sum of the squares of the radii of the ends add one-third of the square of the height; multiply the sum by the height and by 1·5708.

To find the cubic contents of a spheroid, multiply the square of the revolving axis by the fixed axis and by ·5236.

To find the cubic contents of a segment of a spheroid.—When the base is parallel to the revolving axis, multiply the difference between thrice the fixed axis and double the height of the segment by the square of the height, and the product by ·5236.

To find the cubic contents of a wedge.—To twice the length of the base add the length of the edge; multiply the sum by the breadth of the base, and by one-sixth of the height.

To find the cubic contents of a prismoid (a solid of which the two ends are dissimilar, but parallel plane figures of the same number of sides).—To the sum of the areas of the two ends, add four times the area of a section parallel to and equally distant from both ends; and multiply the sum by one-sixth of the length.

To find the surface of a sphere, multiply the square of the diameter by 3·1416.

To find the curve surface of any segment or zone of a sphere, multiply the diameter of the sphere by the height of the zone or segment and by 3·1416.

To find the cubic contents of a sphere, multiply the cube of the diameter by ·5236.

To find the cubic contents of a parabolic conoid, multiply the area of the base by half the height.

To find the cubic contents of a frustum of a parabolic conoid, multiply half the sum of the areas of the two ends by the height of the frustum.

Signification of Signs Used in Calculations.

$=$	signifies	Equality,	as 3 added to $2 = 5$.
$+$	"	Addition,	" $4 + 2 = 6$.
$-$	"	Subtraction,	" $7 - 4 = 3$.
$\times$	"	Multiplication,	" $6 \times 2 = 12$.
$\div$	"	Division,	" $16 \div 4 = 4$.
$: \; : : \; :$	"	Proportion,	" 2 is to 3 so is 4 to 6.
$\sqrt{}$	"	Square Root,	" $\sqrt{16} = 4$.
$\sqrt[3]{}$	"	Cube Root,	" $\sqrt[3]{64} = 4$.
3^2	"	3 is to be squared,	" $3^2 = 9$.
3^3	"	3 is to be cubed,	" $3^3 = 27$.

$2 + 5 \times 4 = 28$ signifies that two, three, or more numbers are to be taken together, as $2 + 5 = 7$, and 4 times $7 = 28$.

$+$, **plus**, means that the number after it is to be *added* to the number before it; thus, $5 + 4$ are 9.

$-$, **minus**, means that the number after it is to be *subtracted* from the number before it; thus, $5 - 4$ is 1.

$\times$, **multiplied by**, means that the number before it is to be *multiplied* by the number after it; thus, 9×3 are 27.

$\div$, **divided by**, means that the number before it is to be *divided* by the number after it; thus, $9 \div 3$ are 3.

$=$, **equal to**, means that the quantity after it is of the same value as the quantity before it; thus, $5 + 6 = 11$.

The Cipher.

The term Cipher has various meanings. It is usually applied to the figure 0, which is equivalent to zero, or nothing. It also means a combination or intertexture of letters, as the initials of a name, the several letters being intertwined so as to form one figure. The word *cipher* also means secret writing; the proper name for which, however, is *cryptogram.*

TABLE

OF DIAMETERS AND AREAS OF SMALL CIRCLES.

DIAM.	AREA.	DIAM.	AREA.	DIAM.	AREA.
·001	·0000008	·027	·0005726	·0625	·0030680
·002	·0000031	·028	·0006158	·065	·0033183
·003	·0000071	·029	·0006605	·070	·0038485
·004	·0000126	·030	·0007069	·075	·0044179
·005	·0000196	·031	·0007548	·080	·0050266
·006	·0000283	·03125	·0007670	·085	·0056745
·007	·0000385	·032	·0008043	·090	·0063617
·008	·0000503	·033	·0008553	·095	·0070882
·009	·0000639	·034	·0009079	·100	·0078540
·010	·0000785	·035	·0009621	·125	·0122719
·011	·0000950	·036	·0010179	·150	·0176715
·012	·0001131	·037	·0010752	·200	·0314159
·013	·0001327	·038	·0011341	·250	·0490875
·014	·0001539	·039	·0011946	·300	·0706858
·015	·0001767	·040	·0012566	·350	·0962115
·015625	·0001917	·041	·0013203	·400	·1256637
·016	·0002016	·042	·0013855	·450	·1590435
·017	·0002270	·043	·0014522	·500	·1963495
·018	·0002545	·044	·0015205	·550	·2375835
·019	·0002835	·045	·0015904	·600	·2827440
·020	·0003142	·046	·0016619	·650	·3318315
·021	·0003464	·047	·0017349	·700	·3848441
·022	·0003801	·048	·0018096	·750	·4417875
·023	·0004155	·049	·0018857	·800	·5026548
·024	·0004524	·050	·0019635	·850	·5674515
·025	·0004909	·055	·0023758	·900	·6361725
·026	·0005309	·060	·0028274	·950	·7088235

TABLE

CONTAINING THE DIAMETERS, CIRCUMFERENCES, AND AREAS OF CIRCLES FROM $\frac{1}{16}$ OF AN INCH TO 100 INCHES, ADVANCING BY $\frac{1}{16}$ OF AN INCH UP TO 10 INCHES, AND BY $\frac{1}{8}$ OF AN INCH FROM 10 TO 100 INCHES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
$\frac{1}{16}$	.1963	.0030	$\frac{7}{16}$	7.6576	4.6664
$\frac{1}{8}$	.3927	.0122	$\frac{1}{2}$	7.8540	4.9087
$\frac{3}{16}$	.5890	.0276	$\frac{9}{16}$	8.0503	5.1573
$\frac{1}{4}$	.7854	.0490	$\frac{5}{8}$	8.2467	5.4119
$\frac{5}{16}$	.9817	.0767	$\frac{11}{16}$	8.4430	5.6727
$\frac{3}{8}$	1.1781	.1104	$\frac{3}{4}$	8.6394	5.9395
$\frac{7}{16}$	1.3744	.1503	$\frac{13}{16}$	8.8357	6.2126
$\frac{1}{2}$	1.5708	.1963	$\frac{7}{8}$	9.0321	6.4918
$\frac{9}{16}$	1.7671	.2485	$\frac{15}{16}$	9.2284	6.7772
$\frac{5}{8}$	1.9635	.3068	3	9.4248	7.0686
$\frac{11}{16}$	2.1598	.3712	$\frac{1}{16}$	9.6211	7.3662
$\frac{3}{4}$	2.3562	.4417	$\frac{1}{8}$	9.8175	7.6699
$\frac{13}{16}$	2.5525	.5185	$\frac{3}{16}$	10.0138	7.9798
$\frac{7}{8}$	2.7489	.6013	$\frac{1}{4}$	10.2120	8.2957
$\frac{15}{16}$	2.9452	.6903	$\frac{5}{16}$	10.4065	8.6179
1	3.1416	.7854	$\frac{3}{8}$	10.6029	8.9462
$\frac{1}{16}$	3.3379	.8861	$\frac{7}{16}$	10.7992	9.2806
$\frac{1}{8}$	3.5343	.9940	$\frac{1}{2}$	10.9956	9.6211
$\frac{3}{16}$	3.7306	1.1075	$\frac{9}{16}$	11.1919	9.9678
$\frac{1}{4}$	3.9270	1.2271	$\frac{5}{8}$	11.3883	10.3206
$\frac{5}{16}$	4.1233	1.3529	$\frac{11}{16}$	11.5846	10.6796
$\frac{3}{8}$	4.3197	1.4848	$\frac{3}{4}$	11.7810	11.0446
$\frac{7}{16}$	4.5160	1.6229	$\frac{13}{16}$	11.9773	11.4159
$\frac{1}{2}$	4.7124	1.7671	$\frac{7}{8}$	12.1737	11.7932
$\frac{9}{16}$	4.9087	1.9175	$\frac{15}{16}$	12.3700	12.1768
$\frac{5}{8}$	5.1051	2.0739	4	12.5664	12.5664
$\frac{11}{16}$	5.3014	2.2365	$\frac{1}{16}$	12.7627	12.9622
$\frac{3}{4}$	5.4978	2.4052	$\frac{1}{8}$	12.9591	13.3640
$\frac{13}{16}$	5.6941	2.5801	$\frac{3}{16}$	13.1554	13.7721
$\frac{7}{8}$	5.8905	2.7611	$\frac{1}{4}$	13.3518	14.1862
$\frac{15}{16}$	6.0868	2.9483	$\frac{5}{16}$	13.5481	14.6066
2	6.2832	3.1416	$\frac{3}{8}$	13.7445	15.0331
$\frac{1}{16}$	6.4795	3.3411	$\frac{7}{16}$	13.9408	15.4657
$\frac{1}{8}$	6.6759	3.5465	$\frac{1}{2}$	14.1372	15.9043
$\frac{3}{16}$	6.8722	3.7582	$\frac{9}{16}$	14.3335	16.3492
$\frac{1}{4}$	7.0686	3.9760	$\frac{5}{8}$	14.5299	16.8001
$\frac{5}{16}$	7.2640	4.2001	$\frac{11}{16}$	14.7262	17.2573
$\frac{3}{8}$	7.4613	4.4302	$\frac{3}{4}$	14.9226	17.7205

TABLE—(Continued)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

Diam.	Circum.	Area.	Diam.	Circum.	Area.
Inch.			Inch.		
13/16	15.1189	18.1900	7/16	23.3656	43.4455
7/8	15.3153	18.6655	1/2	23.5620	44.1787
15/16	15.5716	19.1472	9/16	23.7583	44.9181
5	15.7080	19.6350	5/8	23.9547	45.6636
1/16	15.9043	20.1290	11/16	24.1510	46.4153
1/8	16.1007	20.6290	3/4	24.3474	47.1730
3/16	16.2970	21.1252	13/16	24.5437	47.9370
1/4	16.4934	21.6475	7/8	24.7401	48.7070
5/16	16.6897	22.1661	15/16	24.9364	49.4833
3/8	16.8861	22.6907	8	25.1328	50.2656
7/16	17.0824	23.2215	1/16	25.3291	51.0541
1/2	17.2788	23.7583	1/8	25.5255	51.8486
9/16	17.4751	24.3014	3/16	25.7218	52.8994
5/8	17.6715	24.8505	1/4	25.9182	53.4562
11/16	17.8678	25.4058	5/16	26.1145	54.2748
3/4	18.0642	25.9672	3/8	26.3109	55.0885
13/16	18.2605	26.5348	7/16	26.5072	55.9138
7/8	18.4569	27.1085	1/2	26.7036	56.7451
15/16	18.6532	27.6884	9/16	26.8999	57.5887
6	18.8496	28.2744	5/8	27.0963	58.4264
1/16	19.0459	28.8665	11/16	27.2926	59.7762
1/8	19.2423	29.4647	3/4	27.4890	60.1321
3/16	19.4386	30.0798	13/16	27.6853	60.9943
1/4	19.6350	30.6796	7/8	27.8817	61.8625
5/16	19.8313	31.2964	15/16	28.0780	62.7369
3/8	20.0277	31.9192	9	28.2744	63.6174
7/16	20.2240	32.5481	1/16	28.4707	64.5041
1/2	20.4204	33.1831	1/8	28.6671	65.3968
9/16	20.6167	33.8244	3/16	28.8634	66.2957
5/8	20.8131	34.4717	1/4	29.0598	67.2007
11/16	21.0094	35.1252	5/16	29.2561	68.1120
3/4	21.2058	35.7847	3/8	29.4525	69.0293
13/16	21.4021	36.4505	7/16	29.6488	69.9528
7/8	21.5985	37.1224	1/2	29.8452	70.8823
15/16	21.7948	37.8005	9/16	30.0415	71.8181
7	21.9912	38.4846	5/8	30.2379	72.7599
1/16	22.1875	39.1749	11/16	30.4342	73.7079
1/8	22.3839	39.8713	3/4	30.6306	74.6620
3/16	22.5802	40.5469	13/16	30.8269	75.6223
1/4	22.7766	41.2825	7/8	31.0233	76.5887
5/16	22.9729	41.9974	15/16	31.2196	77.5613
3/8	23.1693	42.7184	10	31.4160	78.5400

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	31.8087	80.5157	3/8	48.3021	185.6612
1/4	32.2014	82.5160	1/2	48.6948	188.6923
3/8	32.5941	84.5409	5/8	49.0875	191.7480
1/2	32.9868	86.5903	3/4	49.4802	194.8282
5/8	33.3795	88.6643	7/8	49.8729	197.9330
3/4	33.7722	90.7627	16	50.2656	201.0624
7/8	34.1649	92.8858	1/8	50.6583	204.2162
11	34.5576	95.0334	1/4	51.0510	207.3946
1/8	34.9503	97.2053	3/8	51.4437	210.5976
1/4	35.3430	99.4021	1/2	51.8364	213.8251
3/8	35.7357	101.6234	5/8	52.2291	217.0772
1/2	36.1284	103.8691	3/4	52.6218	220.3537
5/8	36.5211	106.1394	7/8	53.0145	223.6549
3/4	36.9138	108.4342	17	53.4072	226.9806
7/8	37.3065	110.7536	1/8	53.7999	230.3308
12	37.6992	113.0976	1/4	54.1926	233.7055
1/8	38.0919	115.4660	3/8	54.5853	237.1049
1/4	38.4846	117.8590	1/2	54.9780	240.5287
3/8	38.8773	120.2766	5/8	55.3707	243.9771
1/2	39.2700	122.7187	3/4	55.7634	247.4500
5/8	39.6627	125.1854	7/8	56.1561	250.9475
3/4	40.0554	127.6765	18	56.5488	254.4696
7/8	40.4481	130.1923	1/8	56.9415	258.0161
13	40.8408	132.7326	1/4	57.3342	261.5872
1/8	41.2338	135.2974	3/8	57.7269	265.1829
1/4	41.6262	137.8867	1/2	58.1196	268.8031
3/8	42.0189	140.5007	5/8	58.5123	272.4479
1/2	42.4116	143.1391	3/4	58.9056	276.1171
5/8	42.8043	145.8021	7/8	59.2977	279.8110
3/4	43.1970	148.4896	19	59.6904	283.5294
7/8	43.5897	151.2017	1/8	60.0831	287.2723
14	43.9824	153.9384	1/4	60.4758	291.0397
1/8	44.3751	156.6995	3/8	60.8685	294.8312
1/4	44.7676	159.4852	1/2	61.2612	298.6483
3/8	45.1605	162.2956	5/8	61.6539	302.4894
1/2	45.5532	165.1303	3/4	62.0466	306.3550
5/8	45.9459	167.9896	7/8	62.4393	310.2452
3/4	46.3386	170.8735	20	62.8320	314.1600
7/8	46.7313	173.7820	1/8	63.2247	318.0992
15	47.1240	176.7150	1/4	63.6174	322.0630
1/8	47.5167	179.6725	3/8	64.0101	326.0514
1/4	47.9094	182.6545	1/2	64.4028	330.0643

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
5/8	64.7955	334.1018	7/8	81.2889	525.8375
3/4	65.1882	338.1637	26	81.6816	530.9304
7/8	65.5809	342.2503	1/8	82.0743	536.0477
21	65.7936	346.3614	1/4	82.4670	541.1896
1/8	66.3663	350.4970	3/8	82.8597	546.3561
1/4	66.7590	354.6571	1/2	83.2524	551.5471
3/8	67.1517	358.8419	5/8	83.6451	556.7627
1/2	67.5444	363.0511	3/4	84.0378	562.0027
5/8	67.9371	367.2849	7/8	84.4305	567.2674
3/4	68.3298	371.5432	27	84.8232	572.5566
7/8	68.7225	375.8261	1/8	85.2159	577.8703
22	69.1152	380.1336	1/4	85.6086	583.2085
1/8	69.5079	384.4655	3/8	86.0013	588.5714
1/4	69.9006	388.8220	1/2	86.3940	593.9587
3/8	70.2933	393.2031	5/8	86.7867	599.3706
1/2	70.6860	397.6087	3/4	87.1794	604.8070
5/8	71.0787	402.0388	7/8	87.5721	610.2680
3/4	71.4714	406.4935	28	87.9648	615.7536
7/8	71.8641	410.9728	1/8	88.3575	621.2636
23	72.2568	415.4766	1/4	88.7502	626.7982
1/8	72.6495	420.0049	3/8	89.1429	632.3574
1/4	73.0422	424.5577	1/2	89.5356	637.9411
3/8	73.4349	429.1352	5/8	89.9283	643.5494
1/2	73.8276	433.7371	3/4	90.3210	649.1821
5/8	74.2203	438.3636	7/8	90.7137	654.8395
3/4	74.6130	443.0146	29	91.1064	660.5214
7/8	75.0057	447.6992	1/8	91.4991	666.2278
24	75.3984	452.3904	1/4	91.8918	671.9587
1/8	75.7911	457.1150	3/8	92.2845	677.7143
1/4	76.1838	461.8642	1/2	92.6772	683.4943
3/8	76.5765	466.6380	5/8	93.0699	689.2989
1/2	76.9692	471.4363	3/4	93.4626	695.1280
5/8	77.3619	476.2592	7/8	93.8553	700.9817
3/4	77.7546	481.1065	30	94.2480	706.8600
7/8	78.1473	485.9785	1/8	94.6407	712.7627
25	78.5400	490.8750	1/4	95.0334	718.6900
1/8	78.9327	495.7960	3/8	95.4261	724.6419
1/4	79.3254	500.7415	1/2	95.8188	730.6183
3/8	79.7181	505.7117	5/8	96.2115	736.6193
1/2	80.1108	510.7063	3/4	96.6042	742.6447
5/8	80.5035	515.7255	7/8	96.9969	748.6948
3/4	80.8962	520.7692	31	97.3896	754.7694

TABLE—(Continued)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	97.7823	760.8685	3/8	114.2757	1039.1946
1/4	98.1750	766.9921	1/2	114.6684	1046.3941
3/8	98.5677	773.1404	5/8	115.0611	1053.5281
1/2	98.9684	779.3131	3/4	115.4538	1060.7317
5/8	99.3531	785.5104	7/8	115.8465	1067.9599
3/4	99.7458	791.7322	37	116.2392	1075.2126
7/8	100.1385	797.9786	1/8	116.6319	1082.4898
32	100.5312	804.2496	1/4	117.0246	1089.7915
1/8	100.9240	810.5450	3/8	117.4173	1097.1179
1/4	101.3166	816.8650	1/2	117.8100	1104.4687
3/8	101.7093	823.2096	5/8	118.2027	1111.8441
1/2	102.1020	829.5787	3/4	118.5954	1119.2440
5/8	102.4947	835.9724	7/8	118.9881	1126.6685
3/4	102.8874	842.3905	38	119.3808	1134.1176
7/8	103.2801	848.8333	1/8	119.7735	1141.5911
33	103.6728	855.3006	1/4	120.1662	1149.0892
1/8	104.0655	861.7924	3/8	120.5589	1156.6119
1/4	104.4582	868.3087	1/2	120.9516	1164.1591
3/8	104.8509	874.8497	5/8	121.3443	1171.7309
1/2	105.2436	881.4151	3/4	121.7370	1179.3271
5/8	105.6363	888.0051	7/8	122.1297	1186.9480
3/4	106.0290	894.6196	39	122.5224	1194.5934
7/8	106.4217	901.2587	1/8	122.9151	1202.2633
34	106.8144	907.9224	1/4	123.3078	1209.9577
1/8	107.2071	914.6105	3/8	123.7005	1217.6768
1/4	107.5998	921.3232	1/2	124.0932	1225.4203
3/8	107.9925	928.0605	5/8	124.4859	1233.1884
1/2	108.3852	934.8223	3/4	124.9787	1240.9810
5/8	108.7779	941.6086	7/8	125.2713	1248.7982
3/4	109.1706	948.4195	40	125.6640	1256.6400
7/8	109.5633	955.2550	1/8	126.0567	1264.5062
35	109.9560	962.1150	1/4	126.4494	1272.3970
1/8	110.3487	968.9995	3/8	126.8421	1280.3124
1/4	110.7414	975.9085	1/2	127.2348	1288.2523
3/8	111.1341	982.8422	5/8	127.6275	1296.2168
1/2	111.5268	989.8003	3/4	128.0202	1304.2057
5/8	111.9195	996.7830	7/8	128.4129	1312.2193
3/4	112.3122	1003.7902	41	128.8056	1320.2574
7/8	112.7049	1010.8220	1/8	129.1983	1328.3200
36	113.0976	1017.8784	1/4	129.5910	1336.4071
1/8	113.4903	1024.9592	3/8	129.9837	1344.5189
1/4	113.8830	1032.0646	1/2	130.3764	1352.6551

TABLE—(Continued)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
5/8	130.7691	1360.8159	7/8	147.2625	1725.7324
3/4	131.1618	1369.0012	47	147.6552	1734.9486
7/8	131.5545	1377.2111	1/8	148.0479	1744.1893
42	131.9472	1385.4456	1/4	148.4406	1753.4545
1/8	132.3399	1393.7045	3/8	148.8333	1762.7344
1/4	132.7326	1401.9880	1/2	149.2260	1772.0587
3/8	133.1253	1410.2961	5/8	149.6187	1781.3976
1/2	133.5180	1418.6287	3/4	150.0114	1790.7610
5/8	133.9107	1426.9859	7/8	150.4041	1800.1490
3/4	134.3034	1435.3675	48	150.7968	1809.5616
7/8	134.6961	1443.7738	1/8	151.1895	1818.9986
43	135.0888	1452.2046	1/4	151.5822	1828.4602
1/8	135.4815	1460.6599	3/8	151.9749	1837.9364
1/4	135.8742	1469.1397	1/2	152.3676	1847.4571
3/8	136.2669	1477.6342	5/8	152.7603	1856.9924
1/2	136.6596	1486.1731	3/4	153.1530	1868.5521
5/8	137.0523	1494.7266	7/8	153.5457	1876.1365
3/4	137.4450	1503.3046	49	153.9384	1885.7454
7/8	137.8377	1511.9072	1/8	154.3311	1895.3788
44	138.2304	1520.5344	1/4	154.7238	1905.0367
1/8	138.6231	1529.1860	3/8	155.1165	1914.7093
1/4	139.0158	1537.8622	1/2	155.5092	1924.4263
3/8	139.4085	1546.5530	5/8	155.9019	1934.1579
1/2	139.8012	1555.2883	3/4	156.2946	1943.9140
5/8	140.1939	1564.0382	7/8	156.6873	1953.6947
3/4	140.5866	1572.8125	50	157.0800	1963.5000
7/8	140.9793	1581.6115	1/8	157.4727	1973.3297
45	141.3720	1590.4350	1/4	157.8654	1983.1840
1/8	141.7647	1599.2830	3/8	158.2581	1993.0529
1/4	142.1574	1608.1555	1/2	158.6508	2002.9663
3/8	142.5501	1617.0427	5/8	159.0435	2012.8943
1/2	142.9428	1625.9743	3/4	159.4362	2022.8467
5/8	143.3355	1634.9205	7/8	159.8289	2032.8238
3/4	143.7382	1643.8912	51	160.2216	2042.8254
7/8	144.1209	1652.8865	1/8	160.6143	2052.8515
46	144.5136	1661.9064	1/4	161.0070	2062.9021
1/8	144.9063	1670.9507	3/8	161.3997	2072.9764
1/4	145.2990	1680.0196	1/2	161.7924	2083.0771
3/8	145.6917	1689.1031	5/8	162.1851	2093.2014
1/2	146.0844	1698.2311	3/4	162.5778	2103.3502
5/8	146.4771	1707.3737	7/8	162.9705	2113.5236
3/4	146.8698	1716.5407	52	163.3632	2123.7216

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	163.7559	2133.9440	3/8	180.2493	2585.4509
1/4	164.1486	2144.1910	1/2	180.6423	2596.7287
3/8	164.5413	2154.4626	5/8	181.0347	2608.0311
1/2	164.9340	2164.7587	3/4	181.4274	2619.3580
5/8	165.3267	2175.0794	7/8	181.8201	2630.7095
3/4	165.7194	2185.4245	58	182.2128	2642.0856
7/8	166.1121	2195.7943	1/8	182.6055	2653.4861
53	166.5048	2206.1886	1/4	182.9982	2664.9112
1/8	166.8975	2216.6074	3/8	183.3909	2676.3609
1/4	167.2902	2227.0507	1/2	183.7836	2687.8351
3/8	167.6829	2237.5187	5/8	184.1763	2699.3338
1/2	168.0756	2248.0111	3/4	184 5690	2710.8571
5/8	168.4683	2258.5281	7/8	184.9617	2722.4050
3/4	168.8610	2269.0696	59	185.3544	2733.9774
7/8	169.2537	2279.6357	1/8	185.7471	2745.5743
54	169.6464	2290 2264	1/4	186.1398	2757.1957
1/8	170.0391	2300.8415	3/8	186.5325	2768.8418
1/4	170.4318	2311.4812	1/2	186.9252	2780.5123
3/8	170.8245	2322.1455	5/8	187.3179	2792.2074
1/2	171.2172	2332.8343	3/4	187.7106	2803.9270
5/8	171.6099	2343.5477	7/8	188.1033	2815.6712
3/4	172.0026	2354.2855	60	188.4960	2827.4400
7/8	172.3593	2365.0480	1/8	188.8887	2839.2332
55	172.7880	2375.8350	1/4	189.2814	2851.0510
1/8	173.1807	2386.6465	3/8	189.6741	2862.8934
1/4	173.5734	2397.4825	1/2	190.0668	2874.7603
3/8	173.9661	2408.3432	5/8	190.4595	2886.6517
1/2	174.3588	2419.2283	3/4	190.8522	2898.5677
5/8	174.7515	2430.1833	7/8	191.2419	2910.5083
3/4	175.1442	2441.0772	61	191.6376	2922.4734
7/8	175.5369	2452.0310	1/8	192.0303	2934.4630
56	175.9296	2463.0144	1/4	192.4230	2946.4771
1/8	176.3323	2474.0222	3/8	192.8157	2958.5139
1/4	176.7150	2485.3546	1/2	193.2084	2970.5791
3/8	177.1077	2496.1116	5/8	193.6011	2982.6669
1/2	177.5004	2507.1931	3/4	193.9931	2994.7792
5/8	177.8931	2518.2992	7/8	194.3865	3006.9161
3/4	178.2858	2529.4297	62	194.7792	3019.0776
7/8	178.6785	2543.5849	1/8	195.1719	3031.2635
57	179.0712	2551.7646	1/4	195.5646	3043.4740
1/8	179.4639	2562.9688	3/8	195.9573	3055.7091
1/4	179.8566	2574.1975	1/2	196.3500	3067.9687

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
$\frac{5}{8}$	196.7427	3080.2529	$\frac{7}{8}$	213.2361	3618.3300
$\frac{3}{4}$	197.1354	3092.5615	68	213.6288	3631.6896
$\frac{7}{8}$	197.5281	3104.8948	$\frac{1}{8}$	214.0215	3645.0536
63	197.9208	3117.2526	$\frac{1}{4}$	214.4142	3658.4402
$\frac{1}{8}$	198.3135	3129.6349	$\frac{3}{8}$	214.8069	3671.8554
$\frac{1}{4}$	198.7062	3142.0417	$\frac{1}{2}$	215.1996	3685.2931
$\frac{3}{8}$	199.0989	3154 4732	$\frac{5}{8}$	215.5923	3698.7554
$\frac{1}{2}$	199.4916	3166.9291	$\frac{3}{4}$	215.9850	3712.2421
$\frac{5}{8}$	199.8843	3179.4096	$\frac{7}{8}$	216.3777	3725.7535
$\frac{3}{4}$	200.2770	3191.9146	69	216.7704	3739.2894
$\frac{7}{8}$	200.6697	3204.4442	$\frac{1}{8}$	217.1631	3752.8498
64	201.0624	3216.9984	$\frac{1}{4}$	217.5558	3766.4327
$\frac{1}{8}$	201.4551	3229.5770	$\frac{3}{8}$	217.9485	3780.0443
$\frac{1}{4}$	201.8478	3242.1782	$\frac{1}{2}$	218.3412	3793.6783
$\frac{3}{8}$	202.2405	3254.8080	$\frac{5}{8}$	218.7339	3807.3369
$\frac{1}{2}$	202.6332	3267.4603	$\frac{3}{4}$	219.1266	3821.0200
$\frac{5}{8}$	203.0259	3280.1372	$\frac{7}{8}$	219.5193	3834.7277
$\frac{3}{4}$	203.4186	3292.8385	70	219.9120	3848.4600
$\frac{7}{8}$	203.8113	3305.5645	$\frac{1}{8}$	220.3047	3862.2167
65	204.2040	3318.3151	$\frac{1}{4}$	220.6974	3875.9960
$\frac{1}{8}$	204.5917	3331.0900	$\frac{3}{8}$	221.0901	3889.8039
$\frac{1}{4}$	204.9894	3343.8875	$\frac{1}{2}$	221.4828	3903.6343
$\frac{3}{8}$	205.3821	3356.7137	$\frac{5}{8}$	221.8755	3917.4893
$\frac{1}{2}$	205.7748	3369.5623	$\frac{3}{4}$	222.2682	3931.3687
$\frac{5}{8}$	206.1675	3382.4355	$\frac{7}{8}$	222.6609	3945.2728
$\frac{3}{4}$	206.5602	3395.3332	71	223.0536	3959.2014
$\frac{7}{8}$	206.9529	3408.2555	$\frac{1}{8}$	223.4463	3973.1545
66	207.3456	3421.2024	$\frac{1}{4}$	223.8390	3987.1301
$\frac{1}{8}$	207.7383	3434.1737	$\frac{3}{8}$	224.2317	4001.1344
$\frac{1}{4}$	208.1310	3447.1676	$\frac{1}{2}$	224.6244	4015.1611
$\frac{3}{8}$	208.5237	3468.1901	$\frac{5}{8}$	225.0171	4029.2124
$\frac{1}{2}$	208.9164	3473.2351	$\frac{3}{4}$	225.4098	4043.2882
$\frac{5}{8}$	209.3091	3486.3047	$\frac{7}{8}$	225.8025	4057.3886
$\frac{3}{4}$	209.7018	3499.3987	72	226.1952	4071.5136
$\frac{7}{8}$	210.0945	3512.5174	$\frac{1}{8}$	226.5879	4085.6631
67	210.4872	3525.6606	$\frac{1}{4}$	226.9806	4099.8350
$\frac{1}{8}$	210.8799	3538.8283	$\frac{3}{8}$	227.3733	4114.0356
$\frac{1}{4}$	211.2726	3552.0185	$\frac{1}{2}$	227.7660	4128.2587
$\frac{3}{8}$	211.6653	3565.2374	$\frac{5}{8}$	228.1587	4142.5064
$\frac{1}{2}$	212.0580	3578.4787	$\frac{3}{4}$	228.5514	4156.7785
$\frac{5}{8}$	212.4507	3591.7446	$\frac{7}{8}$	228.9441	4171.0753
$\frac{3}{4}$	212.8434	3605.0350	73	229.3368	4185.3966

TABLE—(Continued)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	229.7295	4199.7424	3/8	246.2229	4824.4299
1/4	230.1222	4214.1107	1/2	246.6156	4839.8311
3/8	230.5149	4228.5077	5/8	247.0083	4855.2568
1/2	230.9076	4242.9271	3/4	247.4010	4870.7071
5/8	231.3003	4257.3711	7/8	247.7937	4886.1820
3/4	231.6930	4271.8396	79	248.1864	4901.6814
7/8	232.0857	4286.3327	1/8	248.5791	4917.2053
74	232.4784	4300.8504	1/4	248.9718	4932.7517
1/8	232.8711	4315.3926	3/8	249.3645	4948.3268
1/4	233.2638	4329.9572	1/2	249.7572	4963.9243
3/8	233.6565	4344.5505	5/8	250.1499	4979.5456
1/2	234.0492	4359.1663	3/4	250.5426	4995.1930
5/8	234.4419	4373.8067	7/8	250.9353	5010.8642
3/4	234.8346	4388.4715	80	251.3280	5026.5600
7/8	235.2273	4403.1610	1/8	251.7207	5042.2803
75	235.6200	4417.8750	1/4	252.1134	5058.0230
1/8	236.0127	4432.6135	3/8	252.5061	5073.7944
1/4	236.4054	4447.3745	1/2	252.8988	5089.5883
3/8	236.7981	4462.1642	5/8	253.2915	5106.4060
1/2	237.1908	4476.9763	3/4	253.6842	5121.2497
5/8	237.5835	4491.8130	7/8	254.0769	5137.1173
3/4	237.9762	4506.6742	81	254.4696	5153.0094
7/8	238.3689	4521.5600	1/8	254.8623	5168.9260
76	238.7616	4536.4704	1/4	255.2550	5184.8651
1/8	239.1543	4551.4023	3/8	255.6477	5200.8329
1/4	239.5470	4566.3626	1/2	256.0404	5216.8231
3/8	239.9397	4581.3486	5/8	256.4331	5232.8371
1/2	240.3324	4596.3571	3/4	256.8258	5248.8772
5/8	240.7251	4611.3902	7/8	257.2185	5264.9411
3/4	241.1178	4626.4477	82	257.6112	5281.0296
7/8	241.5105	4641.3299	1/8	258.0039	5297.1426
77	241.9032	4656.6366	1/4	258.3966	5313.2780
1/8	242.2959	4671.7678	3/8	258.7893	5329.4421
1/4	242.6886	4686.9215	1/2	259.1820	5345.6287
3/8	243.0813	4702.1039	5/8	259.5747	5361.8391
1/2	243.4740	4717.3087	3/4	259.9674	5378.0755
5/8	243.8667	4732.5381	7/8	260.3601	5394.3358
3/4	244.2594	4747.7920	83	260.7528	5410.6206
7/8	244.6521	4763.0705	1/8	261.1455	5426.9299
78	245.0448	4778.3736	1/4	261.5382	5443.2617
1/8	245.4375	4793.7012	3/8	261.9309	5459.6222
1/4	245.8302	4809.0512	1/2	262.3236	5476.0051

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
5/8	262.7163	5492.4118	7/8	279.2097	6203.6905
3/4	263.1090	5508.8446	89	279.6024	6221.1534
7/8	263.5017	5525.3012	1/8	279.9951	6238.6408
84	263.8944	5541.7824	1/4	280.3878	6256.1507
1/8	264.2871	5558.2881	3/8	280.7805	6273.6893
1/4	264.6798	5574.8162	1/2	281.1732	6291.2503
3/8	265.0725	5591.3730	5/8	281.5659	6308.8351
1/2	265.4652	5607.9523	3/4	281.9586	6326.4460
5/8	265.8579	5624.5554	7/8	282.3513	6344.0807
3/4	266.2506	5641.1845	90	282.7440	6361.7400
7/8	266.6433	5657.8357	1/8	283.1367	6379.4238
85	267.0360	5674.5150	1/4	283.5294	6397.1300
1/8	267.4287	5691.2170	3/8	283.9221	6414.8649
1/4	267.8214	5707.9415	1/2	284.3148	6432.6223
3/8	268.2141	5724.6947	5/8	284.7075	6450.4039
1/2	268.6068	5741.4703	3/4	285.1002	6468.2107
5/8	268.9997	5758.2697	7/8	285.4929	6486.0418
3/4	269.3922	5775.0952	91	285.8856	6503.8974
7/8	269.7849	5791.9445	1/8	286.2783	6521.7772
86	270.1776	5808.8184	1/4	286.6710	6539.6801
1/8	270.5703	5825.7168	3/8	287.0637	6557.6114
1/4	270.9630	5842.6376	1/2	287.4564	6573.5651
3/8	271.3557	5859.5871	5/8	287.8491	6593.5431
1/2	271.7484	5876.5591	3/4	288.2418	6611.5462
5/8	272.1411	5893.5549	7/8	288.6345	6629.5736
3/4	272.5338	5910.5767	92	289.0272	6647.6258
7/8	272.9265	5927.6224	1/8	289.4199	6665.7021
87	273.3192	5944.6926	1/4	289.8125	6683.8010
1/8	273.7119	5961.7873	3/8	290.2053	6701.9286
1/4	274.1046	5978.9045	1/2	290.5980	6720.0787
3/8	274.4973	5996.0504	5/8	290.9907	6738.2530
1/2	274.8900	6013.2187	3/4	291.3834	6756.4525
5/8	275.2827	6030.4108	7/8	291.7661	6774.6763
3/4	275.6754	6047.6290	93	292.1688	6792.9248
7/8	276.0681	6064.8710	1/8	292.5615	6811.1974
88	276.4608	6082.1376	1/4	292.9542	6829.4927
1/8	276.8535	6099.4287	3/8	293.3469	6847.8167
1/4	277.2462	6116.7422	1/2	293.7396	6866.1631
3/8	277.6389	6134.0844	5/8	294.1323	6884.5338
1/2	278.0316	6151.4491	3/4	294.5350	6902.9296
5/8	278.4243	6169.8376	7/8	294.9177	6921.3497
3/4	278.8170	6186.2591	94	295.3104	6939.7946

TABLE—(Concluded)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	295.7031	6958.2636	1/8	305.1279	7408.8868
1/4	296.0958	6976.7552	1/4	305.5206	7427.9675
3/8	296.4885	6995.2755	3/8	305.9133	7447.0769
1/2	296.8812	7013.8183	1/2	306.3060	7466.2087
5/8	297.2739	7032.3853	5/8	306.6987	7485.3648
3/4	297.6666	7050.9775	3/4	307.0914	7504.5460
7/8	298.0593	7069.5940	7/8	307.4841	7523.7515
95	298.4520	7088.2352	98	307.8768	7542.9818
1/8	298.8447	7106.9005	1/8	308.2695	7562.2362
1/4	299.2374	7125.5885	1/4	308.6622	7581.5132
3/8	299.6301	7144.3052	3/8	309.0549	7600.8189
1/2	300.0228	7163.0443	1/2	309.4476	7620.1471
5/8	300.4155	7181.8077	5/8	309.8403	7639.4995
3/4	300.8082	7200.5962	3/4	310.2330	7658.8771
7/8	301.2009	7219.4090	7/8	310.6257	7678.2790
96	301.5936	7238.2466	99	311.0184	7697.7056
1/8	301.9863	7257.1083	1/8	311.4111	7717.1563
1/4	302.3790	7275.9926	1/4	311.8038	7736.6297
3/8	302.7717	7294.9056	3/8	312.1965	7756.1318
1/2	303.1644	7313.8411	1/2	312.5892	7775.6563
5/8	303.5571	7332.8008	5/8	312.9819	7795.2051
3/4	303.9498	7351.7857	3/4	313.3746	7814.7790
7/8	304.3425	7370.7949	7/8	313.7673	7834.3772
97	304.7352	7389.8288	100	314.1600	7854.0000

For circumference of circles larger than those given in the table, multiply the diameter by 3,1416.

Example.—Diameter $101'' \times 3{,}1416 = 317{,}3016$.

For areas larger than those in the table, multiply the square of the diameter by the decimal .7854.

Example.—101 inches $\times 101 = 10201 \times .7854 = 8011{,}86$ sq. in.

TABLE

OF LOGARITHMS OF NUMBERS FROM 0 TO 1000.*

No	0	1	2	3	4	5	6	7	8	9	Prop.
0	0	00000	30103	47712	60206	69897	77815	84510	90309	95424	
10	00000	00432	00860	01283	01703	02118	02530	02938	03342	03742	415
11	04139	04532	04921	05307	05690	06069	06445	06818	07188	07554	379
12	07918	08278	08636	08990	09342	09691	10037	10380	10721	11059	349
13	11394	11727	12057	12385	12710	13033	13353	13672	13987	14301	323
14	14613	14921	15228	15533	15836	16136	16435	16731	17026	17318	300
15	17609	17897	18184	18469	18752	19033	19312	19590	19865	20139	281
16	20412	20682	20951	21218	21484	21748	22010	22271	22530	22788	264
17	23045	23299	23552	23804	24054	24303	24551	24797	25042	25285	249
18	25527	25767	26007	26245	26481	26717	26951	27184	27415	27646	236
19	27875	28103	28330	28555	28780	29003	29225	29446	29666	29885	223
20	30103	30319	30535	30749	30963	31175	31386	31597	31806	32014	212
21	32222	32428	32633	32838	33041	33243	33445	33646	33845	34044	202
22	34242	34439	34635	34830	35024	35218	35410	35602	35793	35983	194
23	36173	36361	36548	36735	36921	37106	37291	37474	37657	37839	185
24	38021	38201	38381	38560	38739	38916	39093	39269	39445	39619	177
25	39794	39967	40140	40312	40483	40654	40824	40993	41162	41330	171
26	41497	41664	41830	41995	42160	42324	42488	42651	42813	42975	164
27	43436	43296	43456	43616	43775	43933	44090	44248	44404	44560	158
28	44716	44870	45024	45178	45331	45484	45636	45788	45939	46089	153
29	46240	46389	46538	46686	46834	46982	47129	47275	47421	47567	148
30	47712	47856	48000	48144	48287	48430	48572	48713	48855	48995	143
31	49136	49276	49415	49554	49693	49831	49968	50105	50242	50379	138
32	50515	50650	50785	50920	51054	51188	51321	51454	51587	51719	134
33	51851	51982	52113	52244	52374	52504	52633	52763	52891	53020	130
34	53148	53275	53102	53529	53655	53781	53907	54033	54157	54282	126
35	54407	54530	54654	54777	54900	55022	55145	55266	55388	55509	122
36	55630	55750	55870	55990	56410	56229	56348	56466	56584	56702	119
37	56820	56937	57054	57170	57287	57403	57518	57634	57749	57863	116
38	57978	58002	58206	58319	58433	58546	58658	58771	58883	58995	113
39	59106	59217	59328	59439	59549	59659	59769	59879	59988	60097	110
40	60206	60314	60422	60530	60638	60745	60852	60959	61066	61172	107
41	61278	61384	61489	61595	61700	61804	61909	62013	62117	62221	104
42	62325	62428	62531	62634	62736	62838	62941	63042	63144	63245	102
43	63347	63447	63548	63648	63749	63848	63948	64048	64147	64246	99
44	64345	64443	64542	64640	64738	64836	64933	65030	65127	65224	98
45	65321	65417	65513	65609	65075	65801	65896	65991	66086	66181	96
46	66276	66370	66464	66558	66651	66745	66838	66931	67024	67117	94
47	67210	67302	67394	67486	67577	67669	67760	67851	67942	68033	92
48	68124	68214	68304	68394	68484	68574	68663	68752	68842	68930	90
49	69020	69108	69196	69284	69372	69460	69548	69635	69722	69810	88
50	69897	69983	70070	70156	70243	70329	70415	70500	70586	70671	86
51	70757	70842	70927	71011	71096	71180	71265	71349	71433	71516	84
52	71600	71683	71767	71850	71933	72015	72098	72181	72263	72345	82
53	72428	72509	72591	72672	72754	72835	72916	72997	73078	73158	81
54	73239	73319	73399	73480	73559	73639	73719	73798	73878	73957	80
55	74036	74115	74193	74272	74351	74429	74507	74585	74663	74741	78

* Each logarithm is supposed to have the decimal sign (·) before it.

TABLE — (*Continued.*)

No.	0	1	2	3	4	5	6	7	8	9	Prop.
56	74818	74896	74973	75050	75127	75204	75281	75358	75434	75511	77
57	75587	75663	75739	75815	75891	75966	76042	76117	76192	76267	75
58	76342	76417	76492	76566	76641	76715	76789	76863	76937	77011	74
59	77085	77158	77232	77305	77378	77451	77524	77597	77670	77742	73
60	77815	77887	77959	78031	78103	78175	78247	78318	78390	78461	72
61	78533	78604	78675	78746	78816	78887	78958	79028	79098	79169	71
62	79239	79309	79379	79448	79518	79588	79657	79726	79796	79865	70
63	79934	80002	80071	80140	80208	80277	80345	80413	80482	80550	69
64	80618	80685	80753	80821	80888	80956	81023	81090	81157	81224	68
65	81291	81358	81424	81491	81557	81624	81690	81756	81822	81888	67
66	81954	82020	82805	82151	82216	82282	82347	82412	82477	82542	66
67	82607	82672	82736	82801	82866	82930	82994	83058	83123	83187	65
68	83250	83314	83378	83442	83505	83569	83632	83695	83758	83281	64
69	83884	83947	84010	84073	84136	84198	84260	84323	84385	84447	63
70	84509	84571	84633	84695	84757	84818	84880	84941	85003	85064	62
71	85125	85187	85248	85309	85369	85430	85491	85551	85612	85672	61
72	85733	85793	85853	85913	85973	86033	86093	86153	86213	86272	60
73	86332	86391	86451	86510	86569	86628	86687	86746	86805	86864	59
74	86923	86981	87040	87098	87157	87215	87273	87332	87390	87448	58
75	87506	87564	87621	87679	87737	87794	87852	87909	87966	88024	57
76	88081	88138	88195	88252	88309	88366	88422	88479	88536	88592	56
77	88649	88705	88761	88818	88874	88930	88986	89042	89098	89153	56
78	89209	89265	89320	89376	89431	89487	89542	89597	89652	89707	55
79	89762	89817	89872	89927	89982	90036	90091	90145	90200	90254	54
80	90309	90363	90417	90471	90525	90579	90633	90687	90741	90794	54
81	90848	90902	90955	91009	91062	91115	91169	91222	91275	91328	53
82	91381	91434	91487	91540	91592	91645	91698	91750	91803	91855	53
83	91907	91960	92012	92064	92116	92168	92220	92272	92324	92376	52
84	92427	92479	92531	92582	92634	92685	92737	92788	92839	92890	51
85	92941	92993	93044	93095	93146	93196	93247	93298	93348	93399	51
86	93449	93500	93550	93601	93651	93701	93751	93802	93852	93902	50
87	93951	94001	94051	94101	94151	94200	94250	94300	94349	94398	49
88	94448	94497	94546	94596	94645	94694	94743	94792	94841	94890	49
89	94939	94987	95036	95085	95133	95182	95230	95279	95327	95376	48
90	95424	95472	95520	95568	95616	95664	95712	95760	95808	95856	48
91	95904	95951	95999	96047	96094	96142	96189	96236	96284	96331	48
92	96378	96426	96473	96520	96507	96614	96661	96708	96754	96801	47
93	96848	96895	96941	96988	97034	97081	97127	97174	97220	97266	47
94	97312	97359	97405	97451	97497	97543	97589	97635	97680	97726	46
95	97772	97818	97863	97909	97954	98000	98045	98091	98136	98181	46
96	98227	98272	98317	98362	98407	98452	98497	98542	98587	98632	45
97	98677	98721	98766	98811	98855	98900	98945	98989	99033	99078	45
98	99122	99166	99211	99255	99299	99343	99387	99431	99475	99519	44
99	99563	99607	99657	99694	99738	99782	99825	99869	99913	99956	44

TABLE

OF HYPERBOLIC LOGARITHMS.

NUM.	LOG.	NUM.	LOG.	NUM.	LOG.	NUM.	LOG.
1·01	·0099	1·43	·3576	1·85	·6151	2·27	·8197
1·02	·0198	1·44	·3646	1·86	·6205	2·28	·8241
1·03	·0295	1·45	·3715	1·87	·6259	2·29	·8285
1·04	·0392	1·46	·3784	1·88	·6312	2·30	·8329
1·05	·0487	1·47	·3852	1·89	·6365	2·31	·8372
1·06	·0582	1·48	·3920	1·90	·6418	2·32	·8415
1·07	·0676	1·49	·3987	1·91	·6471	2·33	·8458
1·08	·0769	1·50	·4054	1·92	·6523	2·34	·8501
1·09	·0861	1·51	·4121	1·93	·6575	2·35	·8544
1·10	·0953	1·52	·4187	1·94	·6626	2·36	·8586
1·11	·1043	1·53	·4252	1·95	·6678	2·37	·8628
1·12	·1133	1·54	·4317	1·96	·6729	2·38	·8671
1·13	·1222	1·55	·4382	1·97	·6780	2·39	·8712
1·14	·1310	1·56	·4456	1·98	·6830	2·40	·8754
1·15	·1397	1·57	·4510	1·99	·6881	2·41	·8796
1·16	·1484	1·58	·4574	2·00	·6931	2·42	·8837
1·17	·1570	1·59	·4637	2·01	·6981	2·43	·8878
1·18	·1655	1·60	·4700	2·02	·7030	2·44	·8919
1·19	·1739	1·61	·4762	2·03	·7080	2·45	·8960
1·20	·1823	1·62	·4824	2·04	·7129	2·46	·9001
1·21	·1962	1·63	·4885	2·05	·7178	2·47	·9042
1·22	·1988	1·64	·4946	2·06	·7227	2·48	·9082
1·23	·2070	1·65	·5007	2·07	·7275	2·49	·9122
1·24	·2151	1·66	·5068	2·08	·7323	2·50	·9162
1·25	·2231	1·67	·5128	2·09	·7371	2·51	·9202
1·26	·2341	1·68	·5187	2·10	·7419	2·52	·9242
1·27	·2390	1·69	·5247	2·11	·7466	2·53	·9282
1·28	·2468	1·70	·5306	2·12	·7514	2·54	·9321
1·29	·2546	1·71	·5364	2·13	·7561	2·55	·9360
1·30	·2623	1·72	·5423	2·14	·7608	2·56	·9400
1·31	·2700	1·73	·5481	2·15	·7654	2·57	·9439
1·32	·2776	1·74	·5538	2·16	·7701	2·58	·9477
1·33	·2851	1·75	·5596	2·17	·7747	2·59	·9516
1·34	·2926	1·76	·5653	2·18	·7793	2·60	·9555
1·35	·3001	1·77	·5709	2·19	·7839	2·61	·9593
1·36	·3074	1·78	·5766	2·20	·7884	2·62	·9631
1·37	·3148	1·79	·5822	2·21	·7929	2·63	·9669
1·38	·3220	1·80	·5877	2·22	·7975	2·64	·9707
1·39	·3293	1·81	·5933	2·23	·8021	2·65	·9745
1·40	·3364	1·82	·5988	2·24	·8064	2·66	·9783
1·41	·3435	1·83	·6043	2·25	·8109	2·67	·9820
1·42	·3506	1·84	·6097	2·26	·8153	2·68	·9858

TABLE—(*Continued.*)

Num.	Log.	Num.	Log.	Num.	Log.	Num.	Log.
2·69	·9895	3·11	1·1346	3·53	1·2612	3·95	1·3737
2·70	·9932	3·12	1·1378	3·54	1·2641	3·96	1·3726
2·71	·9969	3·13	1·1410	3·55	1·2669	3·97	1·3787
2·72	1·0006	3·14	1·1442	3·56	1·2697	3·98	1·3812
2·73	1·0043	3·15	1·1474	3·57	1·2725	3·99	1·3837
2·74	1·0079	3·16	1·1505	3·58	1·2753	4·00	1·3862
2·75	1·0116	3·17	1·1537	3·59	1·2781	4·01	1·3887
2·76	1·0152	3·18	1·1568	3·60	1·2809	4·02	1·3912
2·77	1·0188	3·19	1·1600	3·61	1·2837	4·03	1·3937
2·78	1·0224	3·20	1·1631	3·62	1·2864	4·04	1·3962
2·79	1·0260	3·21	1·1662	3·63	1·2892	4·05	1·3987
2·80	1·0296	3·22	1·1693	3·64	1·2919	4·06	1·4011
2·81	1·0331	3·23	1·1724	3·65	1·2947	4·07	1·4036
2·82	1·0367	3·24	1·1755	3·66	1·2974	4·08	1·4060
2·83	1·0402	3·25	1·1786	3·67	1·3001	4·09	1·4085
2·84	1·0438	3·26	1·1817	3·68	1 3029	4·10	1·4109
2·85	1·0473	3·27	1·1847	3·69	1·3056	4·11	1·4134
2 86	1·0508	3·28	1·1878	3·70	1·3083	4·12	1·4158
2·87	1·0543	3·29	1·1908	3·71	1·3110	4·13	1·4182
2·88	1·0577	3·30	1·1939	3·72	1·3137	4·14	1·4206
2·89	1·0612	3·31	1·1969	3·73	1·3164	4·15	1·4231
2·90	1·0647	3·32	1·1999	3·74	1·3190	4·16	1·4255
2·91	1·0681	3·33	1·2029	3·75	1·3217	4·17	1·4279
2 92	1·0715	3·34	1·2059	3·76	1·3244	4·18	1·4303
2·93	1·0750	3·35	1·2089	3·77	1·3271	4·19	1·4327
2·94	1·0784	3·36	1·2119	3·78	1·3297	4·20	1·4350
2·95	1·0818	3·37	1·2149	3·79	1·3323	4·21	1·4374
2·96	1·0851	3·38	1·2178	3·80	1·3350	4·22	1·4398
2·97	1·0885	3·39	1·2208	3·81	1·3376	4·23	1·4421
2·98	1·0919	3·40	1·2237	3·82	1·3402	4·24	1.4445
2·99	1·0952	3·41	1·2267	3·83	1·3428	4·25	1·4469
3·00	1·0986	3·42	1·2296	3·84	1·3454	4·26	1·4492
3·01	1·1019	3·43	1·2325	3·85	1·3480	4·27	1·4516
3·02	1·1052	3·44	1·2354	3·86	1·3506	4·28	1·4539
3·03	1·1085	3·45	1·2387	3·87	1·3532	4·29	1·4562
3·04	1·1118	3·46	1·2412	3·88	1·3558	4·30	1·4586
3·05	1·1151	3·47	1·2441	3·89	1·3584	4·31	1·4609
3·06	1·1184	3·48	1·2470	3·90	1·3609	4·32	1·4632
3·07	1·1216	3·49	1·2499	3·91	1·3635	4·33	1·4655
3·08	1·1249	3·50	1·2527	3·92	1·3660	4·34	1·4678
3·09	1·1281	3·51	1·2556	3·93	1·3686	4·35	1·4701
3·10	1·1314	3·52	1·2584	3·94	1·3711	4·36	1·4724

TABLE—(*Concluded.*)

NUM.	LOG.	NUM.	LOG.	NUM.	LOG.	NUM.	LOG.
4·37	1·4747	4·79	1·5665	5·21	1·6505	5·63	1·7281
4·38	1·4778	4·80	1·5686	5·22	1·6524	5·64	1·7298
4·39	1·4793	4·81	1·5706	5·23	1·6544	5·65	1·7316
4·40	1·4816	4·82	1·5727	5·24	1·6563	5·66	1·7334
4·41	1·4838	4·83	1·5748	5·25	1·6582	5·67	1·7351
4·42	1·4858	4·84	1·5769	5·26	1·6601	5·68	1·7369
4·43	1·4883	4·85	1·5789	5·27	1·6620	5·69	1·7387
4·44	1·4906	4·86	1·5810	5·28	1·6639	5·70	1·7404
4·45	1·4929	4·87	1·5830	5·29	1·6658	5·71	1·7422
4·46	1·4954	4·88	1·5851	5·30	1·6677	5·72	1·7439
4·47	1·4973	4·89	1·5870	5·31	1·6695	5·73	1·7457
4·48	1·4996	4·90	1·5892	5·32	1·6714	5·74	1·7474
4·49	1·5018	4·91	1·5912	5·33	1·6733	5·75	1·7491
4·50	1·5040	4·92	1·5933	5·34	1·6752	5·76	1·7509
4·51	1·5062	4·93	1·5953	5·35	1·6770	5·77	1·7526
4·52	1·5085	4·94	1·5973	5·36	1·6789	5·78	1·7544
4·53	1·5107	4·95	1·5993	5·37	1·6808	5·79	1·7561
4·54	1·5129	4·96	1·6014	5·38	1·6826	5·80	1·7578
4·55	1·5151	4·97	1·6034	5·39	1·6845	5·81	1·7595
4·56	1·5173	4·98	1·6054	5·40	1·6863	5·82	1·7613
4·57	1·5195	4·99	1·6074	5·41	1·6882	5·83	1·7630
4·58	1·5216	5·00	1·6094	5·42	1·6900	5·84	1·7647
4·59	1·5238	5·01	1·6114	5·43	1·6919	5·85	1·7664
4·60	1·5260	5·02	1·6134	5·44	1·6937	5·86	1·7681
4·61	1·5282	5·03	1·6154	5·45	1·6956	5·86	1·7698
4·62	1·5303	5·04	1·6174	5·46	1·6974	5·87	1·7715
4·63	1·5325	5·05	1·6193	5·47	1·6992	5·88	1·7732
4·64	1·5347	5·06	1·6213	5·48	1·7011	5·89	1·7749
4·65	1·5368	5·07	1·6233	5·49	1·7029	5·90	1·7766
4·66	1·5390	5·08	1·6253	5·50	1·7047	5·91	1·7783
4·67	1·5411	5·09	1·6272	5·51	1·7065	5·92	1·7800
4·68	1·5432	5·10	1·6292	5·52	1·7083	5·93	1·7817
4·69	1·5454	5·11	1·6311	5·53	1·7101	5·94	1·7833
4·70	1·5475	5·12	1·6331	5·54	1·7119	5·95	1·7850
4·71	1·5496	5·13	1·6351	5·55	1·7137	5·96	1·7867
4·72	1·5518	5·14	1·6370	5·56	1·7155	5·97	1·7884
4·73	1·5539	5·15	1·6389	5·57	1·7173	5·99	1·7900
4·74	1·5560	5·16	1·6409	5·58	1·7191	6·00	1·7917
4·75	1·5581	5·17	1·6428	5·59	1·7209	6·01	1·7934
4·76	1·5602	5·18	1·6448	5·60	1·7227	6·02	1·7950
4·77	1·5623	5·19	1·6463	5·61	1·7245	6·03	1·7967
4·78	1·5644	5·20	1·6486	5·62	1·7263	6·04	1·7989

Peculiarities of Multiplication.

The multiplication of 987654321 by 45 gives 44444444445. Reversing the order of the digits, and multiplying 123456789 by 45, we get a result equally curious, 5555555505. If we take 123456789 as the multiplicand, and, interchanging the figures 45, take 54 as the multiplier, we obtain another remarkable product, 6666666606. Returning to the multiplicand first used, 987654321, and taking 54 as the multiplier again, we get 53333333334,—all threes except the first and last figures, which read together 54, the multiplier. Taking the same multiplicand, and using 27, the half of 54, as the multiplier, we get a product of 26666666667,—all sixes except the first and last figures, which read together 27, the multiplier. Next, interchanging the figures in the number 27, and using 72 as a multiplier, with 987654321 as the multiplicand, we obtain a product of 71111111112,—all ones except the first and last figures, which read together gives 72, the multiplier.

Decimal Arithmetic.

Decimal Arithmetic is the most simple and explicit mode of performing practical calculations, on account of its doing away with the necessity of fractional parts in the fractional form, thereby reducing long and tedious operations to a few figures.

Decimal Fractions are fractions in which the denominator is a unit, or 1, with ciphers annexed, in which case they are commonly expressed by writing the numerator only with a point before it, by which it is separated from whole numbers; thus ·5, which denotes five-tenths, $\frac{5}{10}$; ·25, that is, $\frac{25}{100}$. Ciphers on the right hand of decimals are of no value whatever; but placed on the left hand, they diminish the decimal value in a tenfold proportion; thus ·6 signifies 6 tenths; ·06 signifies 6 hundredths; and ·006 signifies 6 thousandths of the integer or whole number.

TABLE

OF VULGAR AND DECIMAL FRACTIONS OF AN INCH.

Vulgar Fractions of an Inch.	Decimal Fractions of an Inch.	Vulgar Fractions of an Inch.	Decimal Fractions of an Inch.	Vulgar Fractions of an Inch.	Decimal Fractions of an Inch.
$\frac{1}{32}$	·03125	$\frac{3}{8}$	·375	$\frac{11}{16}$	·6875
$\frac{1}{16}$	·0625	$\frac{3}{8}$ $\frac{1}{32}$	·40625	$\frac{11}{16}$ $\frac{1}{32}$	·71875
$\frac{1}{16}$ $\frac{1}{32}$	·09375	$\frac{7}{16}$	·4375	$\frac{3}{4}$	·75
$\frac{1}{8}$	·125	$\frac{7}{16}$ $\frac{1}{32}$	·46875	$\frac{3}{4}$ $\frac{1}{32}$	·78125
$\frac{1}{8}$ $\frac{1}{32}$	·15625	$\frac{1}{2}$	·5	$\frac{13}{16}$	·8125
$\frac{3}{16}$	·1875	$\frac{1}{2}$ $\frac{1}{32}$	·53125	$\frac{13}{16}$ $\frac{1}{32}$	·84375
$\frac{3}{16}$ $\frac{1}{32}$	·21875	$\frac{9}{16}$	·5625	$\frac{7}{8}$	·875
$\frac{1}{4}$	·25	$\frac{9}{16}$ $\frac{1}{32}$	·59375	$\frac{7}{8}$ $\frac{1}{32}$	·90625
$\frac{1}{4}$ $\frac{1}{32}$	·28125	$\frac{5}{8}$	·625	$\frac{15}{16}$	·9375
$\frac{5}{16}$	·3125	$\frac{5}{8}$ $\frac{1}{32}$	·65625	$\frac{15}{16}$ $\frac{1}{32}$	·96875
$\frac{5}{16}$ $\frac{1}{32}$	·34375				

TABLE.

Common Fraction.	Decimal.	Common Fraction.	Decimal.	Common Fraction.	Decimal.	Common Fraction.	Decimal.
$\frac{1}{32}$	·0312	$\frac{9}{32}$	·2812	$\frac{17}{32}$	·5312	$\frac{25}{32}$	·7812
$\frac{1}{16}$	·0625	$\frac{5}{16}$	·3125	$\frac{9}{16}$	·5625	$\frac{13}{16}$	·8125
$\frac{3}{32}$	·0937	$\frac{11}{32}$	·3437	$\frac{19}{32}$	·5977	$\frac{27}{32}$	·8437
$\frac{1}{8}$	·1250	$\frac{3}{8}$	·3750	$\frac{5}{8}$	·6250	$\frac{7}{8}$	·8750
$\frac{5}{32}$	·1562	$\frac{13}{32}$	·4062	$\frac{21}{32}$	·6562	$\frac{29}{32}$	·9062
$\frac{3}{16}$	·1875	$\frac{7}{16}$	·4375	$\frac{11}{16}$	·6875	$\frac{15}{16}$	·9375
$\frac{7}{32}$	·2187	$\frac{15}{32}$	·4687	$\frac{23}{32}$	·7187	$\frac{31}{32}$	·9687
$\frac{1}{4}$	·2500	$\frac{1}{2}$	·5000	$\frac{3}{4}$	·7500	$\frac{32}{32}$	1·000

Units.

Unit of heat.—The unit of heat varies: the French unit of heat, called a "*caloric,*" is the amount of heat necessary to raise one kilogramme (2·2046215 pounds) of water one degree Centigrade, or from 0° C. to 1° C. In this country and in England the amount of heat necessary to raise one pound of water one degree Fahrenheit, or from 32° Fah. to 33° Fah., is taken as the unit of heat.

For calculations involving quantity of heat, thermometrical temperatures are of no value without a knowledge of the capacity of heat which any body possesses. The quantity of heat required to raise various bodies to any given temperature differs considerably. Water, as possessing the greatest "specific" heat of any known substance, has been universally accepted as a standard, and the unit for the quantity of heat is that amount which will raise 1 pound of water 1° Fah. from a temperature of 32° Fah. To be strictly accurate, the water should be distilled and the lower temperature uniform, in any series of experiments, for the amount of heat to raise water 1° varies slightly at different temperatures.

Unit of length.—The unit of length used in this country and in England is the yard, the length of which has been determined by means of a pendulum vibrating seconds, in the latitude of London, in a vacuum and at the level of the sea. The length of such a pendulum is to be divided into 3,913,929 parts, and 3,600,000 of these parts are to constitute a yard. The yard is divided into 36 inches, so that the length of the seconds pendulum in London is 39·13929 inches.

The division of a foot into 12 inches enables various fractional parts, such as $\frac{1}{6}$, $\frac{1}{4}$, $\frac{1}{3}$, $\frac{1}{2}$, $\frac{2}{3}$, $\frac{3}{4}$, to be made by the use of whole numbers, and, in this respect, it is far more convenient than having the foot divided into ten parts, which will only give $\frac{1}{2}$ and $\frac{1}{5}$ in the whole divisions, without the use of decimals or fractions. So far as the inch is concerned, it is always divided into several

proportions, including tenths, on any good rule, and we use those most preferred, so that it possesses all the advantages of the decimal system with others peculiarly its own.

The French unit of length, called the mètre, has been taken as being the ten-millionth part of the quadrant of a meridian passing through Paris; that is to say, the ten-millionth part of the distance between the equator and the pole, measured through Paris. It is equal to 39·3707898 inches. The mètre is divided into one thousand millimètres, one hundred centimètres, and ten decimètres; while a decimètre is ten mètres, a hectomètre one hundred mètres, a kilomètre one thousand mètres, and a myriamètre ten thousand mètres.

One English yard is equal to 0·91438 mètre; while one mile is equal to 1·60931 kilomètres.

Unit of surface. — For the unit of surface, the square inch, foot, and yard adopted in this country and in England are replaced in the metric system by the square millimètre, centimètre, decimètre, and mètre.

The unit of length squared becomes the unit for surface area, and the same length cubed is the unit for capacity. Cubic inches are generally used to express volumes of water, while cubic feet is a convenient expression for steam.

Unit of capacity. — The cubic inch, foot, and yard furnish measures of capacity; but irregular measures, such as the pint and gallon, are also used in this country and in England. The gallon contains ten pounds avoirdupois weight of distilled water at 62° Fah.; the pint is one-eighth part of a gallon.

The French unit of capacity is the cubic decimètre or litre, equal to 1·7607 English pints, or 0·2200 English gallon; and we have cubic inches, decimètres, centimètres, and millimètres.

Unit of weight. — The unit of weight used in this country and in England, viz., the pound, is derived from the standard gallon, which contains 277·274 cubic inches; the weight of one-tenth of this is the pound avoirdupois, which is divided into 7000 grains.

The French measures of weight are derived at once from the

measures of capacity, by taking the weight of cubic millimètres, centimètres, decimètres, or mètres of water at their maximum density, that is, at 4° C. or 39° Fah.

Unit of time or duration.—The unit of time or duration is the same for all civilized countries. The twenty-fourth part of a mean solar day is called an hour, which contains sixty minutes, which again is divided into sixty seconds. The second is universally used as the unit of duration.

Another unit of time is the period occupied by the earth in making one revolution around the sun, in reference to an assumed fixed star, which unit is called a sidereal year, and contains 365 days, 6 hours, 9 minutes, 9·6 seconds mean solar time.

Unit of velocity.—The units of velocity adopted by different scientific writers vary somewhat; the most usual, perhaps, in regard to sound, falling bodies, projectiles, etc., is the velocity of feet or mètres per second. In the case of light and electricity, miles and kilomètres per second are employed.

Unit of work.—In this country and in England, the unit of work is usually the foot-pound, viz., the force necessary to raise one pound weight one foot above the earth in opposition to the force of gravity. A horse-power is equal to 33,000 pounds raised to a height of one foot in one minute of time.

In France the kilogrammètre is the unit of work, and is the force necessary to raise one kilogramme to a height of one mètre against the force of gravity. One kilogrammètre = 7·233 foot-pounds. The cheval-pouvoir is nearly equal to the English horse-power, and is equivalent to 32,500 pounds raised to a height of one foot in one minute of time. The force competent to produce a velocity of one mètre in one second, in a mass of one gramme, is sometimes adopted as a unit of force.

Unit of pressure.—The pressure of the atmosphere at the level of the ocean, with the barometer at 30 inches, is taken as the unit in estimating and comparing pressures and elastic forces.

TABLE

SHOWING ALL THE UNITS OF LENGTH RECOGNIZED IN ENGLAND SINCE THE SIXTEENTH CENTURY.

3 barleycorns	1 inch.
1·2 inches	1 Surveyor's foot tenth.
1·875 foot tenths (2·25 inches)	1 nail.
1·777 nails (4 inches)	1 hand.
(6·1538 inches side of cube of wine gal. of 223 cub. in.)	
(6·5576 " " " " " beer gal. of 282 cub. in.)	
1·98 hands (7·92 inches)	1 link.
1·136 links (9 inches)	1 quarter.
1·333 quarters (12 inches)	1 foot.
(12·907 inches side of cubic bush. or 2150·42 cub. in.)	
1·875 feet (2·5 quarters)	1 ell Hamburg.
1·2 ell Hamburg (3 quarters)	1 ell Flemish.
1·33 ell Flemish (3 feet)	1 yard.
(38·73 inches side of cubic wine ton of 58212 cub. in.)	
1·25 yards (5 quarters)	1 ell English.
1·644 ell English (6 quarters)	1 ell French.
(5·0397 feet side of cubic cord or 128 cubic feet.)	
1·333 ell French (6 feet)	1 fathom.
2·75 fathoms (5·5 yards)	1 rod, pole or perch.
4 rods (22 yards)	1 chain.
(68·57 yards side of square acre.)	
10 chains	1 furlong.
8 furlongs	1 statute mile.
1·158 statute miles	1 geographical mile.
2·59 geographical miles (3 statute miles)	1 league.

A cubit is two feet.

A great cubit is eleven feet.

A palm is three inches.

A span is ten and seven-eighth inches.

A pace is three feet.

A barrel of flour weighs 196 pounds.

A barrel of pork weighs 200 pounds.

A barrel of powder weighs twenty-five pounds.
A firkin of butter weighs fifty-six pounds.
A tub of butter weighs eighty-four pounds.

Atoms and Molecules.

The term atom has been exclusively appropriated by the chemist, while the mathematician and physicist have preferred to adopt the word molecule to signify those ultimate constituents of matter upon whose motions and relations depend the various states of all bodies solid, liquid, and gaseous. It is said that atoms are attracted to each other by the attraction of cohesion, and repelled by the force of repulsion. By the action of both these forces, the atoms are kept in a state of rest. The solidity of a solid depends upon the fact that each pair of atoms is in this state of equilibrium. These atoms are supposed to be of an oblate, spheroidal form.

The word particle is also freely made use of as involving no hypothesis, and meaning simply a small part of any body. Molecule has been defined by Maxwell as "the smallest possible portion of a particular substance;" and, again, as "that small portion of the substance which moves as one lump in the motion of agitation."

Every substance is now supposed to be composed of an immense number of molecules, which, even in the solid state, are never entirely at rest, and in the gaseous are in a state of perpetual violent commotion, rushing about in straight lines in all directions with inconceivable rapidity.

The difficulty of proving or disproving the molecular theory lies in our inability to determine the size or shape of a molecule by any means in our power. The most powerful microscope fails utterly to show them, and should some material for lenses be discovered infinitely superior to glass or other material at present in use, we should fall far short of appreciating a molecule through the vision.

TABLE

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS OF ALL NUMBERS FROM 1 TO 620.

Number.	Square.	Cube.	Square Root.	Cube Root.
1	1	1	1.	1.
2	4	8	1.4142 136	1.2599 21
3	9	27	1.7230 508	1.4422 496
4	16	64	2.	1.5874 011
5	25	125	2.2360 68	1.7099 759
6	36	216	2.4494 897	1.8171 206
7	49	343	2.6457 513	1.9129 312
8	64	512	2.8284 271	2.
9	81	729	3.	2.0800 837
10	1 00	1 000	3.1622 777	2.1544 347
11	1 21	1 331	3.3166 248	2.2239 801
12	1 44	1 728	3.4641 016	2.2894 286
13	1 69	2 197	3.6055 513	2.3513 347
14	1 96	2 744	3.7416 574	2.4101 422
15	2 25	3 375	3.8729 833	2.4662 121
16	2 56	4 096	4.	2.5198 421
17	2 89	4 913	4.1231 056	2.5712 816
18	3 24	5 832	4.2426 407	2.6207 414
19	3 61	6 859	4.3585 989	2.6684 016
20	4 00	8 000	4.4721 36	2.7144 177
21	4 41	9 261	4.5825 757	2.7589 243
22	4 84	10 648	4.6904 158	2.8020 393
23	5 29	12 167	4.7958 315	2.8438 67
24	5 76	13 824	4.8989 795	2.8844 991
25	6 25	15 625	5.	2.9240 177
26	6 76	17 576	5.0990 195	2.9224 96
27	7 29	19 683	5.1961 524	3.
28	7 84	21 952	5.2915 026	3.0365 889
29	8 41	24 389	5.3851 648	3.0723 168
30	9 00	27 000	5.4772 256	3.1072 325
31	9 61	29 791	5.5677 644	3.1413 806
32	10 24	32 768	5.6568 542	3.1748 021
33	10 89	35 937	5.7445 626	3.2075 343
34	11 56	39 304	5.8309 519	3.2396 118
35	12 25	42 875	5.9160 798	3.2710 663
36	12 96	46 656	6.	3.3019 272
37	13 69	50 653	6.0827 625	3.3322 218
38	14 44	54 872	6.1644 14	3.3619 754
39	15 21	59 319	6.2449 98	3.3912 114
40	16 00	64 000	6.3245 553	3.4199 519
41	16 81	68 921	6.4031 242	3.4482 172

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
42	17 64	74 088	6.4807 407	3.4760 266
43	18 49	79 507	6.5574 385	3.5033 981
44	19 36	85 184	6.6332 496	3.5303 483
45	20 25	91 125	6.7082 039	3.5568 933
46	21 16	97 336	6.7823 3	3.5830 479
47	22 09	103 823	6.8556 546	3.6088 261
48	23 04	110 592	6.9282 032	3.6342 411
49	24 01	117 649	7.	3.6593 057
50	25 00	125 000	7.0710 678	3.6840 314
51	26 01	132 651	7.1414 284	3.7084 298
52	27 04	140 608	7.2111 026	3.7325 111
53	28 09	148 877	7.2801 099	3.7562 858
54	29 16	157 464	7.3484 692	3.7797 631
55	30 25	166 375	7.4161 985	3.8029 525
56	31 36	175 616	7.4833 148	3.8258 624
57	32 49	185 193	7.5498 344	3.8485 011
58	33 64	195 112	7.6157 731	3.8708 766
59	34 81	205 379	7.6811 457	3.8929 965
60	36 00	216 000	7.7459 667	3.9148 676
61	37 21	226 981	7.8102 497	3.9364 972
62	38 44	238 328	7.8740 079	3.9578 915
63	39 69	250 047	7.9372 539	3.9790 571
64	40 96	262 144	8.	4.
65	42 25	274 625	8.0622 577	4.0207 256
66	43 56	287 496	8.1240 384	4.0412 401
67	44 89	300 763	8.1853 528	4.0615 48
68	46 24	314 432	8.2462 113	4.0816 551
69	47 61	328 509	8.3066 239	4.1015 661
70	49 00	343 000	8.3666 003	4.1212 853
71	50 41	357 911	8.4261 498	4.1408 178
72	51 84	373 248	8.4852 814	4.1601 676
73	53 29	389 017	8.5440 037	4.1793 39
74	54 76	405 224	8.6023 253	4.1983 364
75	56 25	421 875	8.6602 54	4.2171 633
76	57 76	438 976	8.7177 979	4.2358 236
77	59 29	456 533	8.7749 644	4.2543 21
78	60 84	474 552	8.8317 609	4.2726 586
79	62 41	493 039	8.8881 944	4.2908 404
80	64 00	512 000	8.9442 719	4.3088 695
81	65 61	531 441	9.	4.3267 487
82	67 24	551 368	9.0553 851	4.3444 815
83	68 89	571 787	9.1104 336	4.3620 707

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
84	70 56	592 704	9.1651 514	4.3795 191
85	72 25	614 125	9.2195 445	4.3968 296
86	73 96	636 056	9.2736 185	4.4140 049
87	75 69	658 503	9.3273 791	4.4310 476
88	77 44	681 472	9.3808 315	4.4479 602
89	79 21	704 969	9.4339 811	4.4647 451
90	81 00	729 000	9.4868 33	4.4814 047
91	82 81	753 571	9.5393 92	4.4979 414
92	84 64	778 688	9.5916 63	4.5143 574
93	86 49	804 357	9.6436 508	4.5306 549
94	88 36	830 584	9.6953 597	4.5468 359
95	90 25	857 375	9.7467 943	4.5629 026
96	92 16	884 736	9.7979 59	4.5788 57
97	94 09	912 673	9.8488 578	4.5947 009
98	96 04	941 192	9.8994 949	4.6104 363
99	98 01	970 299	9.9498 744	4.6260 65
100	1 00 00	1 000 000	10.	4.6415 888
101	1 02 01	1 030 301	10.0498 756	4.6570 095
102	1 04 04	1 061 208	10.0995 049	4.6723 287
103	1 06 09	1 092 727	10.1488 916	4.6875 482
104	1 08 16	1 124 864	10.1980 39	4.7026 694
105	1 10 25	1 157 625	10.2469 508	4.7176 94
106	1 12 36	1 191 016	10.2956 301	4.7326 235
107	1 14 49	1 225 043	10.3440 804	4.7474 594
108	1 16 64	1 259 712	10.3923 048	4.7622 032
109	1 18 81	1 295 029	10.4403 065	4.7768 562
110	1 21 00	1 331 000	10.4880 885	4.7914 199
111	1 23 21	1 367 631	10.5356 538	4.8058 995
112	1 25 44	1 404 928	10.5830 052	4.8202 845
113	1 27 69	1 442 897	10.6301 458	4.8345 881
114	1 29 96	1 481 544	10.6770 783	4.8488 076
115	1 32 25	1 520 875	10.7238 053	4.8629 442
116	1 34 56	1 560 896	10.7703 296	4.8769 99
117	1 36 89	1 601 613	10.8166 538	4.8909 732
118	1 39 24	1 643 032	10.8627 805	4.9048 681
119	1 41 61	1 685 159	10.9087 121	4.9186 847
120	1 44 00	1 728 000	10.9544 512	4.9324 242
121	1 46 41	1 771 561	11.	4.9460 874
122	1 48 34	1 815 848	11.0453 61	4.9596 757
123	1 51 29	1 860 867	11.0905 365	4.9731 898
124	1 53 76	1 906 624	11.1355 287	4.9866 31
125	1 56 25	1 953 125	11.1803 399	5.

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
126	1 58 76	2 000 376	11.2249 722	5.0132 979
127	1 61 29	2 048 383	11.2694 277	5.0265 257
128	1 63 84	2 097 152	11.3137 085	5.0396 842
129	1 66 41	2 146 689	11.3578 167	5.0527 743
130	1 69 00	2 197 000	11.4017 543	5.0657 97
131	1 71 61	2 248 091	11.4455 231	5.0787 531
132	1 74 24	2 299 968	11.4891 253	5.0916 434
133	1 76 89	2 352 637	11.5325 626	5.1044 687
134	1 79 56	2 406 104	11.5758 369	5.1172 299
135	1 82 25	2 460 375	11.6189 5	5.1299 278
136	1 84 96	2 515 456	11.6619 038	5.1425 632
137	1 87 69	2 571 353	11.7046 999	5.1551 367
138	1 90 44	2 628 072	11.7473 401	5.1676 493
139	1 93 21	2 685 619	11.7898 261	5.1801 015
140	1 96 00	2 744 000	11.8321 596	5.1924 941
141	1 98 81	2 803 221	11.8743 421	5.2048 279
142	2 01 64	2 863 288	11.9163 753	5.2171 034
143	2 04 49	2 924 207	11.9582 607	5.2293 215
144	2 07 36	2 985 984	12.	5.2414 828
145	2 10 25	3 048 625	12.0415 946	5.2535 879
146	2 13 16	3 112 136	12.0830 46	5.2656 374
147	2 16 09	3 176 523	12.1243 557	5.2776 321
148	2 19 04	3 241 792	12.1655 251	5.2895 725
149	2 22 01	3 307 949	12.2065 556	5.3014 592
150	2 25 00	3 375 000	12.2474 487	5.3132 928
151	2 28 01	3 442 951	12.2882 057	5.3250 74
152	2 31 04	3 511 008	12.3288 28	5.3368 033
153	2 34 09	3 581 577	12.3693 169	5.3484 812
154	2 37 16	3 652 264	12.4096 736	5.3601 084
155	2 40 25	3 723 875	12.4498 996	5.3716 854
156	2 43 36	3 796 416	12.4899 96	5.3832 126
157	2 46 49	3 869 893	12.5299 641	5.3946 907
158	2 49 64	3 944 312	12.5698 051	5.4061 202
159	2 52 81	4 019 679	12.6095 202	5.4175 015
160	2 56 00	4 096 000	12.6491 106	5.4288 352
161	2 59 21	4 173 281	12.6885 775	5.4401 218
162	2 62 44	4 251 528	12.7279 221	5.4513 618
163	2 65 69	4 330 747	12.7671 453	5.4625 556
164	2 68 96	4 410 944	12.8062 485	5.4737 037
165	2 72 25	4 492 125	12.8452 326	5.4848 066
166	2 75 56	4 574 296	12.8840 987	5.4958 647
167	2 78 89	4 657 463	12.9228 48	5.5068 784

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
168	2 82 24	4 741 632	12.9614 814	5.5178 484
169	2 85 61	4 826 809	13.	5.5287 748
170	2 89 00	4 913 000	13.0384 048	5.5396 583
171	2 92 41	5 000 211	13.0766 968	5.5504 991
172	2 95 84	5 088 448	13.1148 77	5.5612 978
173	2 99 29	5 177 717	13.1529 464	5.5720 546
174	3 02 76	5 268 024	13.1909 06	5.5827 702
175	3 06 25	5 359 375	13.2287 566	5.5934 447
176	3 09 76	5 451 776	13.2664 992	5.6040 787
177	3 13 29	5 545 233	13.3041 347	5.6146 724
178	3 16 84	5 639 752	13.3416 641	5.6252 263
179	3 20 41	5 735 339	13.3790 882	5.6357 408
180	3 24 00	5 832 000	13.4164 079	5.6462 162
181	3 27 61	5 929 741	13.4536 24	5.6566 528
182	3 31 24	6 028 568	13.4907 376	5.6670 511
183	3 34 89	6 128 487	13.5277 493	5.6774 114
184	3 38 56	6 229 504	13.5646 6	5.6877 34
185	3 42 25	6 331 625	13.6014 705	5.6980 192
186	3 45 96	6 434 856	13.6381 817	5.7082 675
187	3 49 69	6 539 203	13.6747 943	5.7184 791
188	3 53 44	6 644 672	13.7113 092	5.7286 543
189	3 57 21	6 751 269	13.7477 271	5.7387 936
190	3 61 00	6 859 000	13.7840 488	5.7488 971
191	3 64 81	6 967 871	13.8202 75	5.7589 652
192	3 68 64	7 077 888	13.8564 065	5.7689 982
193	3 72 49	7 189 057	13.8924 4	5.7789 966
194	3 76 36	7 301 384	13.9283 883	5.7889 604
195	3 80 25	7 414 875	13.9642 4	5.7988 9
196	3 84 16	7 529 536	14.	5.8087 857
197	3 88 09	7 645 373	14.0356 688	5.8186 479
198	3 92 04	7 762 392	14.0712 473	5.8284 867
199	3 96 01	7 880 599	14.1067 36	5.8382 725
200	4 00 00	8 000 000	14.1421 356	5.8480 355
201	4 04 01	8 120 601	14.1774 469	5.8577 66
202	4 08 04	8 242 408	14.2126 704	5.8674 673
203	4 12 09	8 365 427	14.2478 068	5.8771 307
204	4 16 16	8 489 664	14.2828 569	5.8867 653
205	4 20 25	8 615 125	14.3178 211	5.8963 685
206	4 24 36	8 741 816	14.3527 001	5.9059 406
207	4 28 49	8 869 743	14.3874 946	5.9154 817
208	4 32 64	8 998 912	14.4222 051	5.9249 921
209	4 36 81	9 129 329	14.4568 323	5.9344 721

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
210	4 41 00	9 261 000	14.4913 767	5.9439 22
211	4 45 21	9 393 931	14.5258 39	5.9533 418
212	4 49 44	9 528 128	14.5602 198	5.9627 32
213	4 53 69	9 663 597	14.5945 195	5.9720 926
214	4 57 96	9 800 344	14.6287 388	5.9814 24
215	4 62 25	9 938 375	14.6628 783	5.9907 264
216	4 66 56	10 077 696	14.6969 385	6.
217	4 70 89	10 218 313	14.7309 199	6.0092 45
218	4 75 24	10 360 232	14.7648 231	6.0184 617
219	4 79 61	10 503 459	14.7986 486	6.0276 502
220	4 84 00	10 648 000	14.8323 97	6.0368 107
221	4 88 41	10 793 861	14.8660 687	6.0459 435
222	4 92 84	10 941 048	14.8996 644	6.0550 489
223	4 97 29	11 089 567	14.9331 845	6.0641 27
224	5 01 76	11 239 424	14.9666 295	6.0731 779
225	5 06 25	11 390 625	15.	6.0822 02
226	5 10 76	11 543 176	15.0332 964	6.0911 994
227	5 15 29	11 697 083	15.0665 192	6.1001 702
228	5 19 84	11 852 352	15.0996 689	6.1091 147
229	5 24 41	12 008 989	15.1327 46	6.1180 332
230	5 29 00	12 167 000	15.1657 509	6.1269 257
231	5 33 61	12 326 391	15.1986 842	6.1357 924
232	5 38 24	12 487 168	15.2315 462	6.1446 337
233	5 42 89	12 649 337	15.2643 375	6.1534 495
234	5 47 56	12 812 904	15.2970 585	6.1622 401
235	5 52 25	12 977 875	15.3297 097	6.1710 058
236	5 56 96	13 144 256	15.3622 915	6.1797 466
237	5 61 69	13 312 053	15.3948 043	6.1884 628
238	5 66 44	13 481 272	15.4272 486	6.1971 544
239	5 71 21	13 651 919	15.4596 248	6.2058 218
240	5 76 00	13 824 000	15.4919 334	6.2144 65
241	5 80 81	13 997 521	15.5241 747	6.2230 843
242	5 85 64	14 172 488	15.5563 492	6.2316 797
243	5 90 49	14 348 907	15.5884 573	6.2402 515
244	5 95 36	14 526 784	15.6204 994	6.2487 998
245	6 00 25	14 706 125	15.6524 758	6.2573 248
246	6 05 16	14 886 936	15.6843 871	6.2658 266
247	6 10 09	15 069 223	15.7162 336	6.2743 054
248	6 15 04	15 252 992	15.7480 157	6.2827 613
249	6 20 01	15 438 249	15.7797 338	6.2911 946
250	6 25 00	15 625 000	15.8113 883	6.2996 053
251	6 30 01	15 813 251	15.8429 795	6.3079 935

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
252	6 35 04	16 003 008	15.8745 079	6.3163 596
253	6 40 09	16 194 277	15.9059 737	6.3247 035
254	6 45 16	16 387 064	15.9373 775	6.3330 256
255	6 50 25	16 581 375	15.9687 194	6.3413 257
256	6 55 36	16 777 216	16.	6.3496 042
257	6 60 49	16 974 593	16.0312 195	6.3578 611
258	6 65 64	17 173 512	16.0623 784	6.3660 968
259	6 70 81	17 373 979	16.0934 769	6.3743 111
260	6 76 00	17 576 000	16.1245 155	6.3825 043
261	6 81 21	17 779 581	16.1554 944	6.3906 765
262	6 86 44	17 984 728	16.1864 141	6.3988 279
263	6 91 69	18 191 447	16.2172 747	6.4069 585
264	6 96 96	18 399 744	16.2480 768	6.4150 687
265	7 02 25	18 609 625	16.2788 206	6.4231 583
266	7 07 56	18 821 096	16.3095 064	6.4312 276
267	7 12 89	19 034 163	16.3401 346	6.4392 767
268	7 18 24	19 248 832	16.3707 055	6.4473 057
269	7 23 61	19 465 109	16.4012 195	6.4553 148
270	7 29 00	19 683 000	16.4316 767	6.4633 041
271	7 34 41	19 902 511	16.4620 776	6.4712 736
272	7 39 84	20 123 648	16.4924 225	6.4792 236
273	7 45 29	20 346 417	16.5227 116	6.4871 541
274	7 50 76	20 570 824	16.5529 454	6.4950 653
275	7 56 25	20 796 875	16.5831 24	6.5029 572
276	7 61 76	21 024 576	16.6132 477	6.5108 3
277	7 67 29	21 253 933	16.6433 17	6.5186 839
278	7 72 84	21 484 952	16.6783 32	6.5265 189
279	7 78 41	21 717 639	16.7032 931	6.5343 351
280	7 84 00	21 952 000	16.7332 005	6.5421 326
281	7 89 61	22 188 041	16.7630 546	6.5499 116
282	7 95 24	22 425 768	16.7928 556	6.5576 722
283	8 00 89	22 665 187	16.8226 038	6.5654 144
284	8 06 56	22 906 304	16.8522 995	6.5731 385
285	8 12 25	23 149 125	16.8819 43	6.5808 443
286	8 17 96	23 393 656	16.9115 345	6.5885 323
287	8 23 69	23 639 903	16.9410 743	6.5962 023
288	8 29 44	23 887 872	16.9705 627	6.6038 545
289	8 35 21	24 137 569	17.	6.6114 89
290	8 41 00	24 389 000	17.0293 864	6.6191 06
291	8 46 81	24 642 171	17.0587 221	6.6267 054
292	8 52 64	24 897 088	17.0880 075	6.6342 874
293	8 58 49	25 153 757	17.1172 428	6.6418 522

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
294	8 64 36	25 412 184	17.1464 282	6.6493 998
295	8 70 25	25 672 375	17.1755 64	6.6569 302
296	8 76 16	25 934 336	17.2046 505	6.6644 437
297	8 82 09	26 198 073	17.2336 879	6.6719 403
298	8 88 04	26 463 592	17.2626 765	6.6794 2
299	8 94 01	26 730 899	17.2916 165	6.6868 831
300	9 00 00	27 000 000	17.3205 081	6.6943 295
301	9 06 01	27 270 901	17.3493 516	6.7017 593
302	9 12 04	27 543 608	17.3781 472	6.7091 729
303	9 18 09	27 818 127	17.4068 952	6.7165 7
304	9 24 16	28 094 464	17.4355 958	6.7239 508
305	9 30 25	28 372 625	17.4642 492	6.7313 155
306	9 36 36	28 652 616	17.4928 557	6.7386 641
307	9 42 49	28 934 443	17.5214 155	6.7459 967
308	9 48 64	29 218 112	17.5499 288	6.7533 134
309	9 54 81	29 503 609	17.5783 958	6.7606 143
310	9 61 00	29 791 000	17.6068 169	6.7678 995
311	9 67 21	30 080 231	17.6151 921	6.7751 69
312	9 73 44	30 371 328	17.6635 217	6.7824 229
313	9 79 69	30 664 297	17.6918 06	6.7896 613
314	9 85 96	30 959 144	17.7200 451	6.7968 844
315	9 92 25	31 255 875	17.7482 393	6.8040 921
316	9 98 56	31 554 496	17.7763 888	6.8112 847
317	10 04 89	31 855 013	17.8044 938	6.8184 62
318	10 11 24	32 157 432	17.8325 545	6.8256 242
319	10 17 61	32 461 759	17.8605 711	6.8327 714
320	10 24 00	32 768 000	17.8885 438	6.8399 037
321	10 30 41	33 076 161	17.9164 729	6.8470 213
322	10 36 84	33 386 248	17.9443 584	6.8541 24
323	10 43 29	33 698 267	17.9722 008	6.8612 12
324	10 49 76	34 012 224	18.	6.8682 855
325	10 56 25	34 328 125	18.0277 564	6.8753 433
326	10 62 76	34 645 976	18.0554 701	6.8823 888
327	10 69 29	34 965 783	18.0831 413	6.8894 188
328	10 75 84	35 287 552	18.1107 703	6.8964 345
329	10 82 41	35 611 289	18.1383 571	6.9034 359
330	10 89 00	35 937 000	18.1659 021	6.9104 232
331	10 95 61	36 264 691	18.1934 054	6.9173 964
332	11 02 24	36 594 368	18.2208 672	6.9243 556
333	11 08 89	36 926 037	18.2482 876	6.9313 088
334	11 15 56	37 259 704	18.2756 669	6.9382 321
335	11 22 25	37 595 375	18.3030 052	6.9451 496

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
336	11 28 96	37 933 056	18.3303 028	6.9520 533
337	11 35 69	38 272 753	18.3575 598	6.9589 434
338	11 42 44	38 614 472	18.3847 763	6.9658 198
339	11 49 21	38 958 219	18.4119 526	6.9726 826
340	11 56 00	39 304 000	18.4390 889	6.9795 321
341	11 62 81	39 651 821	18.4661 853	6.9863 681
342	11 69 64	40 001 688	18.4932 42	6.9931 906
343	11 76 49	40 353 607	18.5202 592	7.
344	11 83 36	40 707 584	18.5472 37	7.0067 962
345	11 90 25	41 063 625	18.5741 756	7.0135 791
346	11 97 16	41 421 736	18.6010 752	7.0203 49
347	12 04 09	41 781 923	18.6279 36	7.0271 058
348	12 11 04	42 144 192	18.6547 581	7.0338 497
349	12 18 01	42 508 549	18.6815 417	7.0405 806
350	12 25 00	42 875 000	18.7082 869	7.0472 987
351	12 32 01	43 243 551	18.7349 94	7.0540 041
352	12 39 04	43 614 208	18.7616 63	7.0606 967
353	12 46 09	43 986 977	18.7882 942	7.0673 767
354	12 53 16	44 361 864	18.8148 877	7.0740 44
355	12 60 25	44 738 875	18.8414 437	7.0806 988
356	12 67 36	45 118 016	18.8679 623	7.0873 411
357	12 74 49	45 499 293	18.8944 436	7.0939 709
358	12 81 64	45 882 712	18.9208 879	7.1005 885
359	12 88 81	46 268 279	18.9472 953	7.1071 937
360	12 96 00	46 656 000	18.9736 66	7.1137 866
361	13 03 21	47 045 831	19.	7.1203 674
362	13 10 44	47 437 928	19.0262 976	7.1269 36
363	13 17 69	47 832 147	19.0525 589	7.1334 925
364	13 24 96	48 228 544	19.0787 84	7.1400 37
365	13 32 25	48 627 125	19.1049 732	7.1465 695
366	13 39 56	49 027 896	19.1311 265	7.1530 901
367	13 46 89	49 430 863	19.1572 441	7.1595 988
368	13 54 24	49 836 032	19.1833 261	7.1660 957
369	13 61 61	50 243 409	19.2093 727	7.1725 809
370	13 69 00	50 653 000	19.2353 841	7.1790 544
371	13 76 41	51 064 811	19.2613 603	7.1855 162
372	13 83 84	51 478 848	19.2873 015	7.1919 663
373	13 91 29	51 895 117	19.3132 079	7.1984 05
374	13 98 76	52 313 624	19.3390 796	7.2048 322
375	14 06 25	52 734 375	19.3649 167	7.2112 479
376	14 13 76	53 157 376	19.3907 194	7.2176 522
377	14 21 29	53 582 633	19.4164 878	7.2240 45

TABLE—(Continued)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
378	14 28 84	54 010 152	19.4422 221	7.2304 268
379	14 36 41	54 439 939	19.4679 223	7.2367 972
380	14 44 00	54 872 000	19.4935 887	7.2431 565
381	14 51 61	55 306 341	19.5192 213	7.2495 045
382	14 59 24	55 742 968	19.5448 203	7.2558 415
383	14 66 89	56 181 887	19.5703 858	7.2621 675
384	14 74 56	56 623 104	19.5959 179	7.2684 824
385	14 82 25	57 066 625	19.6214 169	7.2747 864
386	14 89 96	57 512 456	19.6468 827	7.2810 794
387	14 97 69	57 960 603	19.6723 156	7.2873 617
388	15 05 44	58 411 072	19.6977 156	7.2936 33
389	15 13 21	58 863 869	19.7230 829	7.2998 936
390	15 21 00	59 319 000	19.7484 177	7.3061 436
391	15 28 81	59 776 471	19.7737 199	7.3123 828
392	15 36 64	60 236 288	19.7989 899	7.3186 114
393	15 44 49	60 698 457	19.8242 276	7.3248 295
394	15 52 36	61 162 984	19.8494 332	7.3310 369
395	15 60 25	61 629 875	19.8746 069	7.3372 339
396	15 68 16	62 099 136	19.8997 487	7.3434 205
397	15 76 09	62 570 773	19.9248 588	7.3495 966
398	15 84 04	63 044 792	19.9499 373	7.3557 624
399	15 92 01	63 521 199	19.9749 844	7.3619 178
400	16 00 00	64 000 000	20.	7.3680 63
401	16 08 01	64 481 201	20.0249 844	7.3741 979
402	16 16 04	64 964 808	20.0499 377	7.3803 227
403	16 24 09	65 450 827	20.0748 599	7.3864 373
404	16 32 16	65 939 264	20.0997 512	7.3925 418
405	16 40 25	66 430 125	20.1246 118	7.3986 363
406	16 48 36	66 923 416	20.1494 417	7.4047 206
407	16 56 49	67 419 143	20.1742 41	7.4107 95
408	16 64 64	67 917 312	20.1990 099	7.4168 595
409	16 72 81	68 417 929	20.2237 484	7.4229 142
410	16 81 00	68 921 000	20.2484 567	7.4289 589
411	16 89 21	69 426 531	20.2731 349	7.4349 938
412	16 97 44	69 934 528	20.2977 831	7.4410 189
413	17 05 69	70 444 997	20.3224 014	7.4470 342
414	17 13 96	70 957 944	20.3469 899	7.4530 399
415	17 22 25	71 473 375	20.3715 488	7.4590 359
416	17 30 56	71 991 296	20.3960 781	7.4650 223
417	17 38 89	72 511 713	20.4205 779	7.4709 991
418	17 47 24	73 034 632	20.4450 483	7.4769 664
419	17 55 61	73 560 059	20.4694 895	7.4829 242

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
420	17 64 00	74 088 000	20.4939 015	7.4888 724
421	17 72 41	74 618 461	20.5182 845	7.4948 113
422	17 80 84	75 151 448	20.5426 386	7.5007 406
423	17 89 29	75 686 967	20.5669 638	7.5066 607
424	17 97 76	76 225 024	20.5912 603	7.5125 715
425	18 06 25	76 765 625	20.6155 281	7.5184 73
426	18 14 76	77 308 776	20.6397 674	7.5243 652
427	18 23 29	77 854 483	20.6639 783	7.5302 482
428	18 31 84	78 402 752	20.6881 609	7.5361 221
429	18 40 41	78 953 589	20.7123 152	7.5419 867
430	18 49 00	79 507 000	20.7364 414	7.5478 423
431	18 57 61	80 062 991	20.7605 395	7.5536 888
432	18 66 24	80 621 568	20.7846 097	7.5595 263
433	18 74 89	81 182 737	20.8086 52	7.5653 548
434	18 83 56	81 746 504	20.8326 667	7.5711 743
435	18 92 25	82 312 875	20.8566 536	7.5769 849
436	19 00 96	82 881 856	20.8806 13	7.5827 865
437	19 09 69	83 453 453	20.9045 45	7.5885 793
438	19 18 44	84 027 672	20.9284 495	7.5943 633
439	19 27 21	84 604 519	20.9523 268	7.6001 385
440	19 36 00	85 184 000	20.9761 77	7.6059 049
441	19 44 81	85 766 121	21.	7.6116 626
442	19 53 64	86 350 888	21.0237 96	7.6174 116
443	19 62 49	86 938 307	21.0475 652	7.6231 519
444	19 71 36	87 528 384	21.0713 075	7.6288 837
445	19 80 25	88 121 125	21.0950 231	7.6346 067
446	19 89 16	88 716 536	21.1187 121	7.6403 213
447	19 98 09	89 314 623	21.1423 745	7.6460 272
448	20 07 04	89 915 392	21.1660 105	7.6517 247
449	20 16 01	90 518 849	21.1896 201	7.6574 138
450	20 25 00	91 125 000	21.2132 034	7.6630 943
451	20 34 01	91 733 851	21.2367 606	7.6687 665
452	20 43 04	92 345 408	21.2602 916	7.6744 303
453	20 52 09	92 959 677	21.2837 967	7.6800 857
454	20 61 16	93 576 664	21.3072 758	7.6857 328
455	20 70 25	94 196 375	21.3307 29	7.6913 717
456	20 79 36	94 818 816	21.3541 565	7.6970 023
457	20 88 49	95 443 993	21.3775 583	7.7026 246
458	20 97 64	96 071 912	21.4009 346	7.7082 388
459	21 06 81	96 702 579	21.4242 853	7.7138 448
460	21 16 00	97 336 000	21.4476 106	7.7194 426
461	21 25 21	97 972 181	21.4709 106	7.7250 325

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
462	21 34 44	98 611 128	21.4941 853	7.7306 141
463	21 43 69	99 252 847	21.5174 348	7.7361 877
464	21 52 96	99 897 344	21.5406 592	7.7417 532
465	21 62 25	100 544 625	21.5638 587	7.7473 109
466	21 71 56	101 194 696	21.5870 331	7.7528 606
467	21 80 89	101 847 563	21.6101 828	7.7584 023
468	21 90 24	102 503 232	21.6333 077	7.7639 361
469	21 99 61	103 161 709	21.6564 078	7.7694 62
470	22 09 00	103 823 000	21.6794 834	7.7749 801
471	22 18 41	104 487 111	21.7025 344	7.7804 904
472	22 27 84	105 154 048	21.7255 61	7.7859 928
473	22 37 29	105 823 817	21.7485 632	7.7914 875
474	22 46 76	106 496 424	21.7715 411	7.7969 745
475	22 56 25	107 171 875	21.7944 947	7.8024 538
476	22 65 76	107 850 176	21.8174 242	7.8079 254
477	22 75 29	108 531 333	21.8403 297	7.8133 892
478	22 84 84	109 215 352	21.8632 111	7.8188 456
479	22 94 41	109 902 239	21.8860 686	7.8242 942
480	23 04 00	110 592 000	21.9089 023	7.8297 353
481	23 13 61	111 284 641	21.9317 122	7.8351 688
482	23 23 24	111 980 168	21.9544 984	7.8405 949
483	23 32 89	112 678 587	21.9772 61	7.8460 134
484	23 42 56	113 379 904	22.	7.8514 244
485	23 52 25	114 084 125	22.0227 155	7.8568 281
486	23 61 96	114 791 256	22.0454 077	7.8622 242
487	23 71 69	115 501 303	22.0680 765	7.8676 13
488	23 81 44	116 214 272	22.0907 22	7.8729 944
489	23 91 21	116 930 169	22.1133 444	7.8783 684
490	24 01 00	117 649 000	22.1359 436	7.8837 352
491	24 10 81	118 370 771	22.1585 198	7.8890 946
492	24 20 64	119 095 488	22.1810 73	7.8944 468
493	24 30 49	119 823 157	22.2036 033	7.8997 917
494	24 40 36	120 553 784	22.2261 108	7.9051 294
495	24 50 25	121 287 375	22.2485 955	7.9104 599
496	24 60 16	122 023 936	22.2710 575	7.9157 832
497	24 70 09	122 763 473	22.2934 968	7.9210 994
498	24 80 04	123 505 992	22.3159 136	7.9264 085
499	24 90 01	124 251 499	22.3383 079	7.9317 104
500	25 00 00	125 000 000	22.3606 798	7.9370 053
501	25 10 01	125 751 501	22.3830 293	7.9422 931
502	25 20 04	126 506 008	22.4053 565	7.9475 739
503	25 30 09	127 263 527	22.4276 615	7.9528 477

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
504	25 40 16	128 024 064	22.4499 443	7.9581 144
505	25 50 25	128 787 625	22.4722 051	7.9633 743
506	25 60 36	129 554 246	22.4944 438	7.9686 271
507	25 70 49	130 323 843	22.5166 605	7.9738 731
508	25 80 64	131 096 512	22.5388 553	7.9791 122
509	25 90 81	131 872 229	22.5610 283	7.9843 444
510	26 01 00	132 651 000	22.5831 796	7.9895 697
511	26 11 21	133 432 831	22.6053 091	7.9947 883
512	26 21 44	134 217 728	22.6274 17	8.
513	26 31 69	135 005 697	22.6495 033	8.0052 049
514	26 41 96	135 796 744	22.6715 681	8.0104 032
515	26 52 25	136 590 875	22.6936 114	8.0155 946
516	26 62 56	137 388 096	22.7156 334	8.0207 794
517	26 72 89	138 188 413	22.7376 340	8.0259 574
518	26 83 24	138 991 832	22.7596 134	8.0311 287
519	26 93 61	139 798 359	22.7815 715	8.0362 935
520	27 04 00	140 608 000	22.8035 085	8.0414 515
521	27 14 41	141 420 761	22.8254 244	8.0466 03
522	27 24 84	142 236 648	22.8473 193	8.0517 479
523	27 35 29	143 055 667	22.8691 933	8.0568 862
524	27 45 76	143 877 824	22.8910 463	8.0620 18
525	27 56 25	144 703 125	22.9128 785	8.0671 432
526	27 66 76	145 531 576	22.9346 899	8.0722 62
527	27 77 29	146 363 183	22.9564 806	8.0773 743
528	27 87 84	147 197 952	22.9782 506	8.0824 8
529	27 98 41	148 035 889	23.	8.0875 794
530	28 09 00	148 877 000	23.0217 289	8.0926 723
531	28 19 61	149 721 291	23.0434 372	8.0977 589
532	28 30 24	150 568 768	23.0651 252	8.1028 39
533	28 40 89	151 419 437	23.0867 928	8.1079 128
534	28 51 56	152 273 304	23.1084 4	8.1129 803
535	28 62 25	153 130 375	23.1300 67	8.1180 414
536	28 72 96	153 990 656	23.1516 738	8.1230 962
537	28 83 69	154 854 153	23.1732 605	8.1281 447
538	28 94 44	155 720 872	23.1948 37	8.1331 87
539	29 05 21	156 590 819	23.2163 735	8.1382 23
540	29 16 00	157 464 000	23.2379 001	8.1432 529
541	29 26 81	158 340 421	23.2594 067	8.1482 765
542	29 37 64	159 220 088	23.2808 935	8.1532 939
543	29 48 49	160 103 007	23.3023 604	8.1583 051
544	29 59 36	160 989 184	23.3238 076	8.1633 102
545	29 70 25	161 878 625	23.3452 351	8.1683 092

TABLE—(Continued)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
546	29 81 16	162 771 336	23.3666 429	8.1733 02
547	29 92 09	163 667 323	23.3880 311	8.1782 888
548	30 03 04	164 566 592	23.4093 998	8.1832 695
549	30 14 01	165 469 149	23.4307 49	8.1882 441
550	30 25 00	166 375 000	23.4520 788	8.1932 127
551	30 36 01	167 284 151	23.4733 892	8.1981 753
552	30 47 04	168 196 608	23.4946 802	8.2031 319
553	30 58 09	169 112 377	23.5159 52	8.2080 825
554	30 69 16	170 031 464	23.5372 046	8.2130 271
555	30 80 25	170 953 875	23.5584 38	8.2179 657
556	30 91 36	171 879 616	23.5796 522	8.2228 985
557	31 02 49	172 808 693	23.6008 474	8.2278 254
558	31 13 64	173 741 112	23.6220 236	8.2327 463
559	31 24 81	174 676 879	23.6431 808	8.2376 614
560	31 36 00	175 616 000	23.6643 191	8.2425 706
561	31 47 21	176 558 481	23.6854 386	8.2474 74
562	31 58 44	177 504 328	23.7065 392	8.2523 715
563	31 69 69	178 453 547	23.7276 21	8.2572 635
564	31 80 96	179 406 144	23.7486 842	8.2621 492
565	31 92 25	180 362 125	23.7697 286	8.2670 294
566	32 03 56	181 321 496	23.7907 545	8.2719 039
567	32 14 89	182 284 263	23.8117 618	8.2767 726
568	32 26 24	183 250 432	23.8327 506	8.2816 255
569	32 37 61	184 220 009	23.8537 209	8.2864 928
570	32 49 00	185 193 000	23.8746 728	8.2913 444
571	32 60 41	186 169 411	23.8956 063	8.2961 903
572	32 71 84	187 149 248	23.9165 215	8.3010 304
573	32 83 29	188 132 517	23.9374 184	8.3058 651
574	32 94 76	189 119 224	23.9582 971	8.3106 941
575	33 06 25	190 109 375	23.9791 576	8.3155 175
576	33 17 76	191 102 976	24.	8.3203 353
577	33 29 29	192 100 033	24.0208 243	8.3251 475
578	33 40 84	193 100 552	24.0416 306	8.3299 542
579	33 52 41	194 104 539	24.0624 188	8.3347 553
580	33 64 00	195 112 000	24.0831 891	8.3395 509
581	33 75 61	196 122 941	24.1039 416	8.3443 41
582	33 87 24	197 137 368	24.1246 762	8.3491 256
583	33 98 89	198 155 287	24.1453 929	8.3539 047
584	34 10 56	199 176 704	24.1660 919	8.3586 784
585	34 22 25	200 201 625	24.1867 732	8.3634 466
586	34 33 96	201 230 056	24.2074 369	8.3682 095
587	34 45 69	202 262 003	24.2280 829	8.3729 668

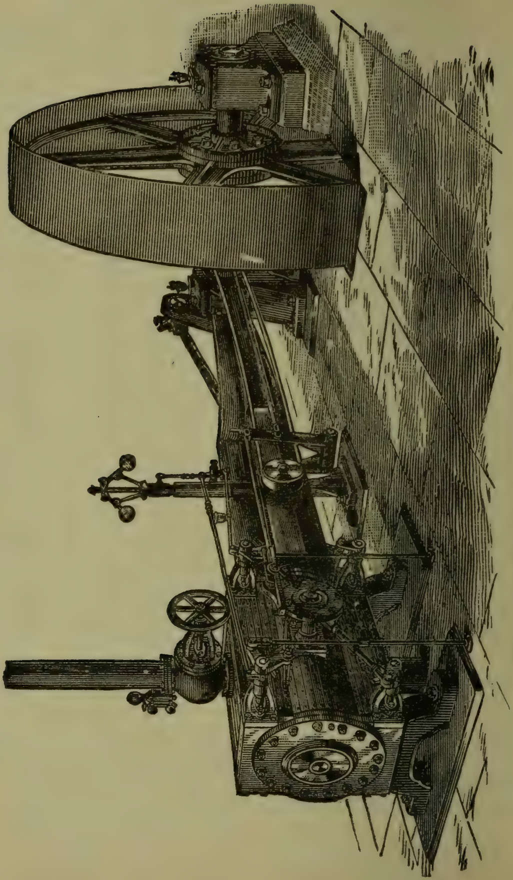
TABLE—(*Concluded*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
588	34 57 44	203 297 472	24.2487 113	8.3777 188
589	34 69 21	204 336 469	24.2693 222	8.3824 653
590	34 81 00	205 379 000	24.2899 156	8.3872 065
591	34 92 81	206 425 071	24.3104 916	8.3919 423
592	35 04 64	207 474 688	24.3310 501	8.3966 729
593	35 16 49	208 527 857	24.3515 913	8.4013 981
594	35 28 36	209 584 584	24.3721 152	8.4061 180
595	35 40 25	210 644 875	24.3926 218	8.4108 326
596	35 52 16	211 708 736	24.4131 112	8.4155 419
597	35 64 09	212 776 173	24.4335 834	8.4202 46
598	35 76 04	213 847 192	24.4540 385	8.4249 448
599	35 88 01	214 921 799	24.4744 765	8.4296 383
600	36 00 00	216 000 000	24.4948 974	8.4343 267
601	36 12 01	217 081 801	24.5153 013	8.4390 098
602	36 24 04	218 167 208	24.5356 883	8.4436 877
603	36 36 09	219 256 227	24.5560 583	8.4483 605
604	36 48 16	220 348 864	24.5764 115	8.4530 281
605	36 60 25	221 445 125	24.5967 478	8.4576 906
606	36 72 36	222 545 016	24.6170 673	8.4623 479
607	36 84 49	223 648 543	24.6373 7	8.467
608	36 96 64	224 755 712	24.6576 56	8.4716 471
609	37 08 81	225 866 529	24.6779 254	8.4762 892
610	37 21 00	226 981 000	24.6981 781	8.4809 261
611	37 33 21	228 099 131	24.7184 142	8.4855 579
612	37 45 44	229 220 928	24.7386 338	8.4901 848
613	37 57 69	230 346 397	24.7588 368	8.4948 065
614	37 69 96	231 475 544	24.7790 234	8.4994 233
615	37 82 25	232 608 375	24.7991 935	8.5040 35
616	37 94 56	233 744 896	24.8193 473	8.5086 417
617	38 06 89	234 885 113	24.8394 847	8.5132 435
618	38 19 24	236 029 032	24.8596 058	8.5178 403
619	38 31 61	237 176 659	24.8797 106	8.5224 321
620	38 44 00	238 328 000	24.8997 992	8.5270 189

Any number multiplied into itself 3 times is cubed; as, $3 \times 3 \times 3 = 27$, which is the cube of 3.

The square root of any number is that number which, multiplied into itself, will be equal to the given number; as, $\sqrt{9} = 3 \times 3$; hence 3 is the square root of 9.



The Wetherill Corliss Steam-Engine.

The Wetherill Corliss Steam-Engine.

The cut on opposite page gives an outline of the general appearance of the Corliss Engine as built by Robert Wetherill & Co., Chester.

The Main Bed is shaped in the strongest form and in direct centre line connecting up cylinder and pedestals. The main pedestal bearings are made in four parts, adjustable. All bearings and wearing surfaces are arranged to take up lost motion occasioned by wear. The proportions, weights, and strength of material are ample. Cylinders are made of hard, strong, charcoal iron, and have all large proportioned port openings, which gives the full boiler pressure against the piston. The cross-head is of an improved pattern, which takes a direct bearing between centre of shoes, and the shoes are gibbed in such a manner that they can be easily removed or any lost motion taken up. Shafts, connecting-rods, and all forgings are made of double-worked hammered iron. Piston-rods, crank-pins, and all other small pins and valve-motion forgings, are of steel. Valve-stems, crank-pin boxes, and valve-gear brasses are all of bronze metal.

The Governor is of the regular Corliss pattern with improvements, but does not require the oil or molasses pot generally used. It acts free under varying loads and pressures, and regulates closely from one horse-power up to the full capacity of engine.

Piston is self-packing, and does not require any attention from the engineer. It keeps the cylinder in good order, requires very little lubrication, and has a reputation of running eight years night and day without any attention, keeping in good order and steam-tight.

Vacuum Dash-Pots for closing valves are generally used. On slow running engines, weights closed with air-cushion are preferred.

Graduating Oil-Cups on all wearing surfaces, and self-feeding oil-cups for cylinders.

Emergencies.

If a follower-plate should break at sea, it might be repaired with boiler-plate and tap-bolts, providing these materials were on board; if not, the propeller-shaft should be detached, and the ship proceed to the nearest port, under sail.

If the air-pump rod should break, and no extra rod be on board the vessel, remove the air-pump bucket and foot-valve, rig a temporary exhaust-pipe with lumber, and proceed to the nearest port.

If a cylinder-head should be fractured or split, it might be repaired temporarily by wrought-iron bars, canvas, or other packing, and tap-bolts.

If the cut-off valve should break at one end, remove it from the other end, and use steam at whole stroke.

If the condenser should become so much out of order as to render it useless, detach the exhaust-pipe from it, and rig a temporary exhaust with such materials as can be found on board.

If the crank-pin or truss-block should heat excessively, allow a stream of water to run on them continually.

If the foot-valve should be rendered useless, the air-pump will work, providing the discharge is in good order. Foot-valves are generally made of vulcanized India-rubber.

If the delivery-pipe should break, burst, or split, it may be repaired temporarily with India-rubber or canvas, lumber, and ropes.

If a crank-pin should break, the broken part may be removed and replaced by a new one, providing there is an extra pin on hand; if not, detach the propeller and proceed under sail.

If the propeller-shaft should twist off, disconnect the engines from it and proceed under sail; but if one or more of the blades should break off, proceed the best way you can, as, while any portion of it remains, it is better than none at all.

Questions,

THE ANSWERS TO WHICH WILL BE FOUND IN THE TEXT.

What is the pressure of the atmosphere at sea-level?

Give the estimated height of the atmosphere.

Give the component parts of atmospheric air.

State the difference in weight between air, water, and mercury.

Does the pressure of the atmosphere differ in different localities?

Is the pressure of the atmosphere constant in the same locality?

Give the altitude of some of the highest mountains in the world.

Give the names of the highest waterfalls in the world.

Give the formula for finding the horse-power of wind-storms.

Give the meaning of the term fuel.

Give the component parts of various kinds of fuel.

Give the comparative values of various kinds of wood for the purpose of fuel as compared with coal.

Give the definitions of the terms fire and smoke.

Define the term heat.

Give the specific heat of different substances.

Give the conductive properties of different substances.

Define the terms combustion and spontaneous combustion.

Give the component parts of fresh water.

Is the specific gravity of all waters the same?

Give the latent heat of water, ice, and steam.

Define the term vapor.

Give the meaning of the term gases.

What is meant by the term area?

Give the rules for finding the diameters, circumferences, and areas of circles.

Give the meaning of the term cipher.

Give the meaning of the terms atoms and molecules.

What advantages does decimal arithmetic possess?

What are decimal fractions?

Give an explanation of the different recognized units, such as those of heat, length, surface, capacity, weight, time, velocity, work, and pressure.

Give the different lengths which have been recognized in England since the sixteenth century.

Demonstrate the difference between vulgar and decimal fractions.

Give the decimals for the 16th or 32d part of an inch.

PART EIGHTH.

Lexicon of Definitions of Central, Mechanical, and Dynamical Forces.

Acceleration.— Acceleration is the increase of velocity in a moving body, caused by the continued action of the motive force. When bodies in motion pass through equal spaces in equal time, or, in other words, when the velocity of the body is the same during the period that the body is in motion, it is termed uniform motion, of which we have a familiar instance in the motion of the hands of a clock over the face of it; but a more correct illustration is the revolution of the earth on its axis. In the case of a body moving through unequal spaces in equal times, or with a varying velocity, if the velocity increase with the duration of the motion, it is termed accelerated motion; but, if it decrease with the duration of the motion, it is termed retarded motion.

Affinity.— Affinity is a term used in chemistry to denote that kind of attraction, by which the particles of different bodies unite, and form a compound, possessing properties distinct from those of any of the substances which compose it. Thus, when an acid and alkali combine, a new substance is formed called a salt, perfectly different in its chemical properties from either an acid or an alkali; and, in consequence of the law of affinity, these bodies have a tendency to unite.

Angle.— If two lines, drawn on a plain surface, are so situated that they meet in a point, or would do so, if sufficiently prolonged, they form an opening, which is called an angle. One straight line, meeting another which is perpendicular to it, makes the angle on both sides equal; then these angles are each called a right angle, and, in this case, the one line is said to be perpendicular to the other, or, in the language of mechanics, the one line is said to be square with the other; and if the one line be horizontal, the perpendicular is said to be plumb to it. The arc,

which measures a right angle, is the quarter of the whole circumference, or a quadrant, and contains 90 degrees; any angle measured by an arc less than this is acute (sharp), and if by an arc greater than a quadrant, obtuse (blunt).

Axle.— An axle is a shaft supporting a wheel; the wheel may turn on the axle, or be fastened to it, and the axle turn on bearings. Axles are viewed as having certain relations to girders in principle. Girders generally have their two ends resting on two points of support, and the load is either located at fixed distances from the props, or dispersed over the whole surface of the axle; the wheels may be considered the props, and the journals the loaded parts. It is found that the inclined surface of the wheel-tire, given by coning ranges from 1 to 12 to 1 to 20; and, as a matter of course, the direct tendency of the wheel under a load is to descend that incline, so that every vertical blow, which the wheels may receive, is compounded of two forces, viz., the one to crush the wheels in the direction of their vertical plane, and the other to move the lower parts of the wheels together. It will be seen that these two forces have a direct tendency to bend the axle somewhere between the wheels.

Attraction.— The terms attraction, or affinity, and repulsion, in the language of modern scientists, are employed merely as the expression of one of two general facts, either that the masses or particles of matter have a tendency to approach and unite to, or to recede from, one another, under certain circumstances. The term attraction is used synonymously with affinity. All bodies have a tendency to attract each other, more or less, and it is this power which is called attraction. Attraction is mutual; it extends to indefinite distances. All bodies, whatever, as well as their component elementary particles, are endued with it. It is not annihilated, at however great a distance we suppose them to be placed from each other; neither does it disappear, though they be arranged ever so near each other. The nature of this reciprocal attraction, or at least the cause which produces it, is altogether unknown to us. Whether it be inherent in all matter, or whether it be the conse-

quence of some other agent, are questions beyond the reach of human understanding; but its existence is nevertheless certain.

Capillary Attraction.— Capillary attraction is the property inherent in narrow tubes and porous substances, such as sponge, lamp-wicking, thread, etc., of raising oil, water, or other fluids above their natural level. Hence this principle is applied for obtaining a continuous supply of lubricating fluids between rubbing and revolving surfaces in motion, by means of a siphon constructed of wickings, worsted, or some other substance, one end of which is immersed in oil, and the other inserted in the tube through which the fluid is to be conducted.

Centre of Gravity. — The forces with which all bodies tend to fall to the earth may be considered parallel; hence every body may be considered as acted on by a system of parallel forces whose resultant may be found, and these forces, in all positions of the body, act on the same points in the same vertical direction. There is, therefore, in every body a point through which the resultant always passes, in whatever position it is placed. This point is called the centre of gravity of the body. The centre of gravity of a uniform cylinder or prism is in its axis, and at the middle of its length; of a right cone or a pyramid it is also in the axis, but at one-fourth of the height from the base.

Cohesion. — Cohesion is that quality of the particles of a body which causes them to adhere to each other, and to resist being torn apart.

Dynamics. — Dynamics is that branch of mechanics which treats of forces in motion producing power and work. It comprehends the action of all kinds of machinery, manual and animal labor, in the transformation of physical work.

Elastic Fluids. — Elastic fluids are divided into two classes — permanent gases and vapors. The gases cannot be converted into the liquid state by any known practicable process; whereas the vapors are readily reduced to the liquid form by pressure or diminution of temperature. In respect of their mechanical properties, there is, however, no essential difference between the two classes.

Elastic fluids, in a state of equilibrium, are subject to the action of two forces, namely, gravity, and a molecular force acting from particle to particle.

Elasticity. — Elasticity is that quality which enables a body to return to its original form, after having been distorted or stretched by some external force. The limit of elasticity is the extent to which any material may be stretched without receiving a permanent set.

Energy. — This term has become obsolete in a mechanical point of view, and is now only applied to the action of men and animals. If an individual man, horse, ox, or other animal performed a certain amount of work in less time than another would occupy in doing the same work, we say that he acted with great energy; but when a machine runs fast, or fire burns fast, or the waves roll fast, we do not apply the term energy, but simply say that the machine runs at a very high speed, or increased its speed; or the fire burned fiercely, or that the wind blew, or the waves rolled with great violence.

Force. — Force is the cause of motion or change of motion in material bodies. Every change of motion, viz., every change in the velocity of a body, must be regarded as the effect of a force. On the other hand, rest, or the invariability of the state of motion of a body, must not be attributed to the absence of forces, since opposite forces destroy each other and produce no effect. The force of gravity with which a body falls to the ground, still acts, though the body rests; but this action is counteracted by the solidity of the material upon which it reposes. Forces that are balanced so as to produce rest, are called statical forces or pressures, to distinguish them from moving, deflecting, accelerating, or retarding forces, *i. e.*, such as are producing motion, or a change in the direction or velocity of motion. This distinction is wholly artificial, as the same force may act in any of these modes; it may sometimes be a statical and sometimes an accelerating force.

Force is any action which can be expressed simply by weight, and is distinguished by a great variety of terms, such as attraction,

repulsion, gravity, pressure, tension, compression, cohesion, adhesion, resistance, inertia, strain, stress, strength, thrust, burden, load, squeeze, pull, push, pinch, punch, etc., all of which may be measured or expressed by weight without regard to motion, time, power, or work.

Focus. — Focus in geometry is that point in the transverse axis of a conic section, at which the double ordinate is equal to a perimeter, or to a third proportional to the transverse and conjugate axis.

Friction. — Friction is the resistance offered to the motion of a body, when pressed upon the surface of another body which does not partake of its motion. Under these circumstances, the surfaces in contact have a certain tendency to adhere. Not being perfectly smooth, the imperceptible asperities which may be supposed to exist on all surfaces, however highly polished, become to some extent interlocked, and, in consequence, a certain amount of force is requisite to overcome the mutual resistance to motion of the two surfaces, and to maintain the sliding motion even when it has been produced. By increasing the pressure, the resistance to motion is increased also; and on the other hand, by rendering the surfaces smoother by lubrication, its amount is greatly diminished, but can never be entirely annulled.

Friction cannot be strictly called a force, unless that term be taken in a negative sense. The tendency of force, in the rigid meaning of the word, is to produce motion; whereas the tendency of friction is to destroy motion.

Friction Rollers. — The obstruction which a cylinder meets in rolling along a smooth plane, is quite distinct in its character, and far inferior in its amount, to that which is produced by the friction of the same cylinder drawn lengthwise along a plane. For example, in the case of wood rolling on wood, the resistance is to the pressure, if the cylinder be small, as 16 or 18 to 1000, and if the cylinder be large, this may be reduced to 6 to 1000. The friction from sliding, in the same cases, would be to the pressure as 2 to 10 or 3 to 10, according to the nature of the wood. Hence,

by causing one body to roll on another, the resistance is diminished from 12 to 20 times. It is therefore a principle, in the composition of machines, that attrition should be avoided as much as possible, and rolling motions substituted whenever circumstances permit.

Gravity and Gravitation.— These terms are often used synonymously, to denote the mutual tendency which all bodies in nature have to approach each other. Gravity acts on gases in the same manner as on all other material substances; but the action of the molecular forces is altogether different from that which takes place among the elementary particles of solids and liquids, as in the case of solid bodies, the molecules strongly attract each other (hence results their cohesion), and, in the case of liquids, exert a feeble or evanescent attraction, so as to be indifferent to internal motion; but, in the case of the gases, the molecular forces are repulsive, and the molecules, yielding to the action of these forces, tend incessantly to recede from each other, and, in fact, do recede, until their further separation is prevented by an exterior obstacle.

Gravity, Specific.— The specific gravity of a body is the ratio of its weight to an equal volume of some other body assumed as a conventional standard. The standard usually adopted for solids and liquids is rain or distilled water at a common temperature. In bodies of equal magnitudes, the specific gravities are directly as the weights or as their densities. In bodies of the same specific gravity the weights will be as the magnitudes. In bodies of equal weights, the specific gravities are inversely as the magnitudes. The weights of different bodies are to each other in the compound ratio of their magnitudes and specific gravities. Hence, it is obvious that, speaking of the magnitude, weight, and specific gravity of a body, if any two of them are given, the third may be found. A body, immersed in a fluid, will sink if its specific gravity be greater than that of the fluid; if it be less, the body will rise to the top, and be only partly immersed; and, if the specific gravity of the body and fluid be equal, it will remain at rest in any part of the fluid in which it may be placed. When a body is heavier

than a fluid, it loses as much of its weight when immersed as is equal to a quantity of the fluid of the same bulk or magnitude. If the specific gravity of the fluid be greater than that of the body, then the quantity of fluid displaced by the part immersed is equal to the weight of the whole body. And hence, as the specific gravity of the fluid is to that of the body, so is the whole magnitude of the body to the part immersed. The specific gravities of equal solids are as their parts immersed in the same fluid.

Gyration, the Centre of.— The centre of gyration is that point in which, if all the matter contained in a revolving system were collected, the same angular velocity will be generated in the same time by a given force acting at any place, as would be generated by the same force acting similarly in the body or system itself. The distance of the centre of gyration from the point of suspension or the axis of motion, is a mean proportional between the distances of the centres of oscillation and gravity from the same point or axle.

Horse-power, or Power of a Horse.— The power of a horse when applied to draw loads, as well as when made the standard of comparison for determining the value of other powers, has been variously stated. The relative strength of men and horses depends, of course, upon the manner in which their strength is applied. Thus, the worst way of applying the strength of a horse is to make him carry a weight up a steep hill. The power of a horse varies from five to eleven times that of a man.

Hydrodynamics.— Hydrodynamics is that branch of general mechanics which treats of the equilibrium and motion of fluids. The terms hydrostatics and hydrodynamics have a signification corresponding to the terms statics and dynamics in the mechanics of solid bodies, viz., hydrostatics is that division of the science which treats of the equilibrium of fluids, and hydrodynamics that which relates to their forces and motion. It is, however, usual to include the whole doctrine of the mechanics of fluids under the general term of hydrodynamics, and to denote the divisions rela-

tive to their equilibrium and motion by the terms hydrostatics and hydraulics.

Hyperbola.—A plane figure, formed by cutting a section from a cone by a plane parallel to its axis, or to any plane within the cone, which passes through the cone's vertex. The curve of the hyperbola is such, that the difference, between the distances of any point in it from two given points, is always equal to a given right line. If the vertices of two cones meet each other, so that their axes form one continuous straight line, and the plane of the hyperbola cut from one of the cones be continued, it will cut the other cone, and form what is called the opposite hyperbola, equal and similar to the former; and the distance, between the vertices of the two hyperbolæ, is called the major axis, or transverse diameter. If the distance between a certain point within the hyperbola, called the focus, and any point in the curve be subtracted from the distance of said point in the curve from the focus of the opposite hyperbola, the remainder will always be equal to a given quantity, that is, to the major axis; and the distance of either focus from the centre of the major axis is called the eccentricity. The line passing through the centre, perpendicular to the major axis, and having the distance of its extremities from those of the axis equal to the eccentricity, is called the minor axis, or conjugate diameter. An ordinate to the major axis, a double ordinate, and an absciss mean the same as the corresponding lines in the parabola.

Impact is the single instantaneous blow or stroke communicated from one body in motion to another either in motion or at rest.

Impenetrability.—In physics, one of the essential properties of matter or body. It is a property inferred from invariable experience, and resting on this incontrovertible fact, that no two bodies can occupy the same portion of space in the same instant of time. Impenetrability, as respects solid bodies, requires no proof: it is obvious to the touch. With regard to liquids, the property may be proved by very simple experiments. Let a vessel be filled to

the brim with water, and a solid, incapable of solution in water, be plunged into it; a portion of the water will overflow, exactly equal in bulk to the dimensions of the body immersed. If a cork be rammed hard into the neck of a vial full of water, the vial will burst, while its neck remains entire. The disposition of air to resist penetration may be illustrated in the following way: Let a tall glass vessel be nearly filled with water, on the surface of which a lighted taper is set to float; if, over this glass, a smaller cylindrical vessel, likewise of glass, be inverted and pressed downwards, the contained air maintaining its place, the internal body of the water will descend, while the rest will rise up at the sides, and the taper will continue to burn for some seconds encompassed by the whole mass of liquid.

Impetus.— Impetus is the product of the mass and velocity of a moving body, considered as instantaneous, as distinguished from momentum, with reference to time, and from force, with reference to capacity of continuing its motion. Impetus in gunnery is the altitude through which a heavy body must fall to acquire a velocity equal to that with which the ball is discharged from the piece.

Incidence.— The term incidence in mechanics is used to denote the direction in which a body or ray of light strikes another body, and is otherwise called inclination. In moving bodies, their incidence is said to be perpendicular or oblique, according as their lines of motion make a straight line or an angle at the point of contact.

Inclination.— Inclination denotes the mutual approach or tendency of two bodies, lines, or planes towards each other, so that the lines of their direction make at the point of contact an angle of greater or less magnitude.

The Inclined Plane.— The inclined plane is the representative of the second class of mechanical powers. Its fundamental law of action is that of the composition and resolution of forces. The manner in which the advantage is immediately derived from it, is therefore distinct from that of the first class; there is necessarily

a fulcrum, a point round which all the motion takes place, and through which the power acts on the resistance; whereas, in this class, there is no apparent centre of action. The advantage gained by the inclined plane, when the power acts in a parallel direction to the plane, is as the length to the height or angle of inclination. Hence, divide the weight by the ratio of inclination, and the quotient equals the power that will support that weight upon the plane. Or, multiply the weight by the height of the plane and divide by the length; the quotient is the power.

The descent of a body down an inclined plane is as the length of the plane to its height; so is the velocity acquired by a falling body through a given height to the velocity on an inclined plane.

Ex.—A body will roll down an inclined plane 300 feet long and 25 feet high in one second of time, as follows: 300 : 25 : : 16 : 1·33 = the distance which the body descends per second on an in-inclined plane.

Inertia.—Inertia is that property of matter by which it tends, when at rest, to remain so, and, when in motion, to continue in motion.

Levers.—Levers are classified into three different kinds or orders. When the fulcrum is between the force and the weight, the lever is called a lever of the first order; when the weight is between the force and the fulcrum, the lever is of the second order; when the force is between the weight and the fulcrum, the lever is of the third order. The levers of safety-valves for steam-boilers belong to this last class.

The lever is an inflexible bar, by the application of which one force may balance or overcome another. These forces are termed, respectively, the *power* and the *resistance* or *weight*, not from any difference in the action of the forces, but with reference merely to the intention with which the machine is used; and, indeed, the same terms are used about all the other mechanical elements. In applying the rod to operate upon any resistance, it must rest upon a centre prop, or fulcrum, somewhere along its length, upon which

it turns in the performance of its work. Thus, there are three points in every lever to be regarded in examining its action, namely, the two points of application of the power, the weight, and the point resting on the fulcrum. There is a certain relation to be observed between the magnitudes of the opposing force and the distances from the fulcrum, namely, that in every case the power *multiplied* by its distance from the fulcrum is equal to the weight *multiplied* by its distance from the same point. From this relation, simple rules may be deduced for calculation.

To know the power to be applied, at a certain distance from the fulcrum, to overcome a resistance acting also at a certain distance, *multiply* the resistance by its distance from the fulcrum, which gives its momentum, and divide the product by the distance given; the quotient will be the power, it being understood that the distance and the force be each expressed in the same unit of measure. For example, a weight, 1120 lbs., at 3 inches from the fulcrum, is to be balanced by a force at the distance of 10 feet. Now, 10 feet are equal to 120 inches; and the momentum of 1120 lbs. is $1120 \times 3 = 3360$. Divide this by 120, we have 28 lbs. for the power required. Again, To know the distance at which a given force ought to be applied to balance a given weight at a certain distance, we must, in like manner, multiply the weight by its distance, as before, and divide by the given power. 1120 lbs., for example, at 3 inches distance, are to be balanced by a force of 28 lbs. To find the distance of this weight, 1120 lbs. multiplied by 3 gives 3360, which, divided by 28, gives 120 inches, or 10 feet.

Machines.— Machines are instruments employed to regulate motion, so as to save either time or force. The maximum effect of machines is the greatest effect which can be produced by them. In all machines that work with a uniform motion, there is a certain velocity, and a certain load of resistance, that yields the greatest effect, and which are therefore more advantageous than any other. A machine may be so heavily charged, that the motion, resulting from the application of any given power, will be but just sufficient to overcome it, and, if any motion ensue, it

will be very trifling, and the whole effect will be very slight. If the machine is very lightly loaded, it may give great velocity to the load; but, from the smallness of its quantity, the effect may still be very inconsiderable; consequently, between these two loads, there must be some intermediate one that will render the effect the greatest possible. This is equally true in the application of animal strength, as in machines. The maximum effect of a machine is produced when the weight or resistance to be overcome is four-ninths of that which the power, when fully exerted, is able to balance, or of that resistance which is necessary to reduce the machine to rest; and the velocity of the part of the machine, to which the power is applied, should be one-third of the greatest velocity of the power.

The moving power and the resistance being both given, if the machine be so constructed, that the velocity of the point, to which the power is applied, be to the velocity of the point to which the resistance is applied, as four times the resistance to nine times the power, the machine will work to the greatest possible advantage. This is equally true when applied to the strength of animals; that is, a man, horse, or other animal, will do the greatest quantity of work, by continued labor, when his strength is opposed to a resistance equal to four-ninths of his natural strength, and his velocity equal to one-third of his greatest velocity when not impeded. In all machines, simple as well as compound, what is gained in power is lost in time; but the loss of time is compensated by convenience. The power of a machine is not altered by varying the size of the wheels, provided the proportion, produced by the multiplication of the power of the several parts, remains the same.

Mechanics.— Mechanics is that branch of natural philosophy which treats of three simple physical elements, force, motion, and time, with their combinations, constituting power, space, and work. Mechanics, regarded as a science, comprehends the sum of our knowledge relative to the sensible motions of bodies either actually existing, or expressed by the opposition of forces tend-

ing to produce motion. The science is thus resolvable into a code of discovered laws, applying to the causes which occasion and modify the direction and the velocities of motion, and is therefore distinct from those branches of science, in which, although presenting phenomena of motion in sensible portions of matter, we do not consider the circumstances and laws of these motions, but only the effects produced.

When motion itself is considered, the reasoning belongs to mechanics, and it is probable that, as our knowledge of the laws which govern the phenomena that are evolved under the hand of the experimental philosopher becomes more extended, a wider meaning will be given to the science of motion. The definition which is here given of mechanics is not coeval with the name. The science, like most other sciences, has gradually expanded to its present extent. It was originally the science of machines — these being the first subjects of its speculation; and, as every material combination employed for producing or preventing motion may be regarded as a machine, and may be resolved into the same elementary principles as those employed in machines,— the mechanical powers,— the name "mechanics" came to be applied to motion, and the tendency to motion of any bodies whatever. Mechanics still continues to be defined by some the "science of force," and there does not appear to be any valid objection to the definition. Force is the cause of motion, and its laws are identical with the laws of motion; and, consequently, the science of force coincides, in all its parts, with the science of motion, which is mechanics.

All machinery, when analyzed, will be found to consist of a combination of six simple machines or elements, commonly called *mechanical powers.* The six elements are respectively the *lever,* the *pulley,* the *wheel* and *axle,* the *inclined plane,* the *wedge* and the *screw.* Though they are not powers, or, in other words, sources of power or force, yet they transmit and diffuse or concentrate forces. The essential idea of machinery is, that it renders force available for effecting practical ends. Machines

prepare, as it were, the raw material of force supplied to us from natural sources. It is transmitted and modified by certain combinations of the elements of machinery, and is given off, at last, in a condition suitable for producing the desired mechanical effect. We do not create force; the object of machinery is to transmit it, and diffuse or concentrate it in one or more points of action. The various diffused or concentrated forces, then, being added together, will amount exactly to the original available force.

Modulus.— The modulus of the elasticity of any substance is a column of the same substance, capable of producing a pressure on its base, which is to the weight causing a certain degree of compression, as the length of the substance is to the diminution in its length.

Momentum.— Momentum, in mechanics, is the same as impetus or quantity of motion, and is generally estimated by the product of the velocity and the mass of the body. This is a subject which has led to various controversies between philosophers,— some estimating it by the mass into the velocity as stated above, while others maintain that it varies as the mass into the square of the velocity. But this difference seems to have arisen rather from a misconception of the term than from any other cause. Those who maintain the former doctrine, understand momentum to signify the momentary impact; and the advocates of the latter doctrine recognize it as the sum of all the impulses till the motion of the body is destroyed.

The momentum of a body is the power contained in a moving body, and is equal to its weight multiplied by its velocity.

The momentum divided by the velocity equals the weight, or the momentum divided by the weight equals the velocity.

The velocity acquired by a falling body is proportional to the time; or the velocity acquired at the end of the first second, multiplied by the number of seconds, will be the velocity with which it strikes the ground.

The space through which a body falls in a given time may be

found by multiplying the square of seconds by the distance which a body moves in one second from a state of rest, which is $16\frac{1}{12}$ feet, or 193 inches, and the product will be the whole space through which a body falls in a certain time; if multiplied in feet, the product will be feet, and if in inches, the product will be inches.

The space through which a body falls in any number of seconds may be calculated as follows: During the first second, a body falls $16\frac{1}{12}$ feet; the second second, it will fall 3 times $16\frac{1}{12}$ feet; and the third second, 5 times $16\frac{1}{12}$ feet.

The distance passed over by a body in an air-tight vessel by the force of gravity is $16\frac{1}{12}$ ft.; it gradually acquires an accelerated motion, so that it has a velocity of $32\frac{3}{12}$ ft. at the end of the first second.

Ex.—If a substance weighing 336 pounds be dropped from a height of 400 feet, its momentum, and the time it takes to reach the ground, may be calculated as follows:

$$16 : 1 : : \sqrt{400} = 5 \text{ seconds, the time of falling.}$$

Now, to get the momentum, we must have the velocity to multiply into the weight, and 5 seconds being the time it was falling, $1 : 32 : : 5 : \; 160 =$ velocity in feet $\times$ 336 (weight) $=$ 53,760 pounds momentum.

Ex.—If a ball 24 pounds in weight be dropped from a height of 400 feet, the velocity with which it will strike the ground, and its momentum, may be thus calculated. The time of falling must be first found. Then, $16 : 1 : : \sqrt{400} = \sqrt{25} = 5$ seconds is the time of falling. Since the velocity is proportional to the time, $1 : 5 : : 32 : 160$, which is the velocity in feet with which it strikes the ground. Then, as the momentum is equal to the velocity multiplied by the weight, we have $24 \times 160 = 3840$, the momentum.

Motion.—Motion, in mechanics, is a change of place, or it is that property inherent in matter by which it passes from one point of space to another. Absolute motion is the absolute change of place in a moving body, independent of any other motion whatever; in which general sense, however, it never falls

under our observation. All those motions, which we consider as absolute, are in fact only relative, being referred to the earth, which is itself in motion. By absolute motion, therefore, we must only understand that which is so with regard to some fixed point upon the earth, this being the sense in which it is interpreted by writers on this subject. Accelerated motion is that which is continually receiving constant accessions of velocity. Angular motion is the motion of a body as referred to a centre, about which it revolves. Compound motion is that which is produced by two or more powers acting in different directions. Natural motion is that which is natural to bodies, or that which arises from the action of gravity. Parallel Motions.—Contrivances of this kind are required for the conversion of rotary and alternating angular motion into rectilineal motion, and the converse; but the absolute necessity there is of guiding the path of a piston in a steam-engine, has called forth more attention to the principles and mechanism of parallel motions than would otherwise, in all probability, have been awarded to the subject. Relative motion is the relative change of place in one or more moving bodies. Retarded motion is that which suffers continual diminution of velocity, the laws of which are the reverse of those of accelerated motion. Rotary motion, turning as a wheel on its axis, pertaining to or resembling the motion of a wheel. Rotary motions were favorite ones with ancient philosophers. They considered a circle as the most perfect of all figures, and erroneously concluded that a body in motion would naturally revolve in one.

To the substitution of circular for straight motions, and of continuous for alternating ones, may be attributed nearly all the conveniences and elegancies of civilized life. It is not too much to assert, that the present advanced state of science and the arts is due to revolving mechanism. From the earliest times it had been an object to convert, whenever practicable, the rectilinear and reciprocating movements of machines into circular and continuous ones. Old mechanics seem to have been led to this result by that tact or natural sagacity, that is more or less common to

all times and people. Thus the dragging of heavy loads on the ground led to the adoption of wheels and rollers,—hence carts and carriages. The rotary movements of the drill superseded the alternating one of the punch and gouge, in making perforations; the whetstone gave way to the revolving grindstone; the turning-lathe produced round forms infinitely more accurate and more expeditiously than the uncertain and irregular carving or cutting with the knife. Motion is uniform, when a body moves continually with the same velocity, passing over equal spaces in equal times.

Oscillation, Centre of.—The centre of oscillation is that point in a vibrating body, in which, if the whole were concentrated and attached to the same axis of motion, it would vibrate in the same time the body does in its natural state. The centre of oscillation is situated in a right line passing through the centre of gravity, and perpendicular to the axis of motion.

Pendulum.—If any heavy body, suspended by an inflexible rod from a fixed point, be drawn aside from the vertical position, and then let fall, it will describe the arc of a circle, of which the point of suspension is the centre. On reaching the vertical position, it will have acquired a velocity equal to that which it would have acquired by falling vertically through the versed sine of the arc which it has described, in consequence of which it will continue to move in the same arc until the whole velocity is destroyed; and, if no other force than gravity were in operation, this would take place, when the body reached a height on the opposite side of the vertical height, equal to that from which it fell. Having reached this height, it would again descend, and so continue to vibrate forever; but, in consequence of the friction of the axis and the resistance of the air, each successive vibration will be diminished, and the body soon be brought to rest in the vertical position. A body thus suspended and caused to vibrate is called a pendulum; and the passage from the greatest distance from the vertical on the one side to the greatest distance on the other is called an oscillation.

Percussion.—The centre of percussion is that point in a body

revolving about an axis at which, if it struck an immovable obstacle, all its motion would be destroyed, or it would not incline either way. When an oscillating body vibrates with a given angular velocity, and strikes an obstacle, the effect of the impact will be the greatest, if it be made at the centre of percussion; since, in this case, the obstacle receives the whole revolving motion of the body; whereas, if the blow be struck at any other point, a part of the motion will be employed in endeavoring to continue the rotation.

Perpetual Motion.— In mechanics, a machine which, when set in motion, would continue to move forever, or, at least, until destroyed by the friction of its parts, without the aid of any exterior cause, would constitute perpetual motion. The discovery of perpetual motion has always been a celebrated problem in mechanics, on which many ingenious, though in general ill-instructed, persons have spent their time; but all the labor bestowed on it has proved abortive. In fact, the impossibility of its existence has been fully demonstrated from the known laws of matter. In speaking of perpetual motion, it is to be understood that, from among the forces by which motion may be produced, we are to exclude not only air and water, but other natural agents, as heat, atmospheric changes, etc. The only admissible agents are the inertia of matter, and its attractive forces, which may all be considered of the same kind as gravitation. It is an admitted principle in philosophy, that action and reaction are equal, and that, when motion is communicated from one body to another, the first loses just as much as is gained by the second. But every moving body is continually retarded by two passive forces,— the resistance of the air and friction. In order, therefore, that motion may be continued without diminution, one of two things is necessary — either that it be maintained by an exterior force, (in which case it would cease to be what we understand by a perpetual motion,) or that the resistance of the air and friction be annihilated, which is practically impossible.

The motion cannot be perpetuated, till these retarding forces

are compensated, and they can only be compensated by an exterior force, as the force, communicated to any body, cannot be greater than the generating force, which is only sufficient to continue the same quantity of motion, when there is no resistance. The error, of confounding mere pressure with energy available to produce power, is the main origin of the majority of attempts at perpetual motion, and even sometimes causes, among confused minds, exaggerated expectations about the effects to be obtained from mechanical contrivances. A wound-up spring is exactly equivalent to a weight. It may exert a certain pressure, great in proportion to its size and strength; but, unless it is allowed to unwind it, it cannot produce motion or power. It is the same with compressed air or gases; they are, in fact, nothing but wound-up springs, with this difference, however, that, in place of needing mechanical power to wind them up, we may use either heat, chemical agencies, or electricity.

Pneumatics. — Pneumatics is the science which treats of the mechanical properties of elastic fluids, and particularly of atmospheric air. Elastic fluids are divided into two classes — permanent gases and vapors. The gases cannot be converted into the liquid state by any known process; whereas the vapors are readily reduced to the liquid form by pressure or diminution of temperature. In respect to their mechanical properties, there is, however, no essential difference between the two classes. Elastic fluids, in a state of equilibrium, are subject to the action of two forces, namely, gravity, and a molecular force acting from particle to particle. Gravity acts on the gases in the same manner as on all other substances; but the action of the molecular forces is altogether different from that which takes place among the elementary particles of solids and liquids; for, in the case of solid bodies, the molecules strongly attract each other (whence results their cohesion), and, in the case of liquids, exert a feeble or evanescent attraction, so as to be indifferent to internal motion; but, in the case of the gases, the molecular forces are repulsive, and the molecules, yielding to the action of these forces, tend incessantly to

recede from each other, and, in fact, do recede until their further separation is prevented by an exterior obstacle. Thus air, confined within a close vessel, exerts a constant pressure against the interior surface, which is not sensible, only because it is balanced by the equal pressure of the atmosphere on the exterior surface. This pressure exerted by the air against the sides of a vessel, within which it is confined, is called its elasticity — its elastic force or tension.

Power. — Power is the product of force and velocity; that is to say, a force multiplied by the velocity with which it is acting. The term horse-power is a unit of power, equivalent to a force of 33,000 pounds acting with a velocity of one foot per minute, or 150 pounds acting with a velocity of 220 feet per minute, which is the same as a force of 550 pounds acting with a velocity of one foot per second. Man-power is a unit of power established by Morin to be equivalent to 50 foot-pounds of power, or 50 effects; that is to say, a man turning a crank with a force of 50 pounds, with a velocity of one foot per second, is a standard man-power.

Power implies the ability to do so much work *in a certain time*, and, like other things which we talk about and compare, requires a unit by which to measure it. The unit used in this country is called a horse-power, and is equal to raising 33,000 lbs. through a space of one foot in a minute of time, or in any other way performing 33,000 foot-pounds of work in a minute.

Pressure. — Pressure is force acting against some obstacle or opposing force. It differs from weight, inasmuch as pressure exerts a force in all directions, whereas weight exerts its influence only in one. There are instances where weight causes pressure in more than one direction, *e. g.*, in fluids, while there are others in which pressure has no connection with weight, such as the pressure of steam in a boiler.

Prime Movers.—Prime movers are those machines from which we obtain power through their adaptation to the transformation of some available natural force into that kind of effort which develops mechanical power.

The Pulley.—Pulleys are of two kinds, *fixed* and *movable.* The fixed pulley only turns upon its axis, and affords no mechanical advantage; therefore, when the power and the weight are equal, they balance each other. It is used for the convenience of changing the direction of a motion. The *movable* pulley not only turns upon its axis, but rises and falls with its weight. Every movable pulley may be considered as hanging by two ropes equally stretched, and which, consequently, are equal portions of the weight; therefore each pulley of this sort doubles the power. The principle of the pulley, as practically applied in the block and tackle, is the distribution of weight on various points of support, the mechanical advantage derived depending entirely upon the flexibility and tension of the rope and the number of pulleys or sheaves in the lower or rising block. Hence, by blocks and tackle of the usual kind, the power is to the weight as the number of cords attached to the lower block. The advantages to be gained by the employment of the block and tackle may be found by dividing the weight to be raised by the number of cords leading to, from, or attached to the lower block, and the quotient is the power required to produce an equilibrium, provided friction does not exist. Or, divide the weight to be raised by the power to be applied; the quotient is the number of sheaves in, or cords attached to, the rising block.

The Screw.—The screw is another modification of the inclined plane, and it may be said to remove the same kind of practical inconveniences incidental to the use of the latter, that the pulley does in reference to the simple lever. The lever is very limited in the extent of its action; so is the inclined plane. But the pulley multiplies the extent of the action of the lever, by presenting, in effect, a series of levers acting in regular succession; and just such a purpose is effected by the screw. It multiplies the extent of the action of the inclined plane by presenting, in effect, a continued series of planes.

The screw, in principle, is that of an inclined plane wound round a cylinder, which generates a spiral of uniform inclination, each revolution producing a rise or traverse motion equal to the

pitch of the screw or distance between the two consecutive threads, the pitch being the height or angle of inclination, and the circumference the length of the plane. Hence, the mechanical advantage is as the circumference of the circle described by the lever where the power acts is to the pitch of the screw, so is the force to the resistance in principle.

To find the effective power obtained by a screw of $\frac{7}{8}$-inch pitch, and moved by a force equal to 50 lbs. at the extremity of a lever 30 inches in length:

$$\frac{30 \times 2 \times 3{\cdot}1416 \times 50}{875} = 10{,}760 \text{ lbs.}$$

To find the power necessary to overcome a resistance equal to 7000 lbs. by a screw of $1\frac{1}{4}$-inch pitch, and moved by a lever 25 inches in length:

$$\frac{7000 \times 1{\cdot}25}{25 \times 2 \times 3{\cdot}1416} = 55{\cdot}73 \text{ lbs.}$$

In the case of a screw acting upon the periphery of a toothed wheel, the power is to the resistance as the product of the circle's circumference described by the winch or lever and radius of the wheel to the product of the screw's pitch and radius of the axle, or point whence the power is transmitted; but observe that, if the screw consist of more than one thread, the apparent pitch must be increased so many times as there are threads in the screw. Hence, to find what weight a given power will equipoise, multiply together the radius of the wheel, the length of the lever at which the power acts, the magnitude of the power, and the constant number 6·2832; divide the product by the radius of the axle into the pitch of the screw, and the quotient is the weight that the power is equal to.

Resilience is a characteristic of bodies, which manifests a certain degree of flexibility before they can be broken, hence the body that bends or yields the most at the time of fracture is the toughest.

Statics is the science of forces in equilibrium. It treats of the strength of materials, of bridges, and of girders; the stability of

walls, steeples, and towers; the static momentum of levers, with their combinations into weighing-scales, windlasses, pulleys, funicular machines, inclined planes, screws, catenaria, and all kinds of gearing.

Strength.—Strength is the resistance which a body opposes to a disintegration or separation of its parts. *Tensile* strength is the absolute resistance which a body makes to being torn apart by two forces acting in opposite directions. *Crushing* strength is the resistance which a body opposes to being battered or flattened down by any weight placed upon it. *Transverse* strength is the resistance to bending, or flexure, as it is called. *Torsional* strength is the resistance which a body offers to any external force which attempts to twist it round. *Detrusive* strength is the resistance which a body offers to being clipped or shorn into two parts by such instruments as shears or scissors. *Working* strength. The term "working strength" implies a certain reduction made in the estimate of the strength of materials, so that, when the instrument or machine is put to use, it may be capable of resisting a greater strain than it is expected on the average to sustain.

Tools.—By the term tools, according to the definition given by Rennie, we understand instruments employed in the manual arts for facilitating mechanical operations, by means of percussion, penetration, separation, and abrasion, of the substances operated upon, and for all which operations various motions are required to be imparted either to the tool or to the work.

Torsion.—Torsion, in mechanics, is the twisting or wrenching of a body by the exertion of a lateral force. If a slender rod of metal, suspended vertically, and having its upper end fixed, be twisted through a certain angle by a force acting in a plane perpendicular to its axis, it will, on the removal of the force, untwist itself, or return in the opposite direction with a greater or less velocity, and, after a series of oscillations, will come to rest in its original position. The limits of torsion, within which the body will return to its original state, depend on its elasticity. A fine wire of a few feet in length may be twisted through several revo-

lutions, without impairing its elasticity; and, within those limits, the force evolved is found to be perfectly regular, and directly proportional to the angular displacement from the position of rest. If the angular displacement exceeds a certain limit (as in a wire of lead, for example, before disruption takes place), the particles will assume a new arrangement, or take a set, and will not return to their original position on the withdrawal of the disturbing force.

Velocity.—Velocity is the rate of motion. Velocity is independent of space and time, but, in order to obtain its value or expression as a quantity, we compare space with time. Thus, when the value of the velocity of a moving body is required, we measure the space which the body passes through, and divide that space by the time of passage, and the quotient is the velocity.

Weight.—The weight of a body is the force of attraction between the earth and that body. The weight of a body is greatest at the surface of the earth, and decreases above or below that surface. Above the surface, the weight decreases as the square of its distance from the centre of the earth, and below the surface the weight decreases simply as its distance from the centre.

Weights and Measures.—The weights and measures of this country are identical with those of England. In both countries they repose, in fact, upon actually existing masses of metal (brass), which have been individually declared by law to be the units of the system. In scientific theory, they are supposed to rest upon a permanent and universal law of nature — the gravitation of distilled water at a certain temperature and under a certain atmospheric pressure. In this aspect, the origination is with the grains, which must be such that 252,458 of these units of brass will be in just equilibrium with a cubic inch of distilled water, when the mercury stands at 30 inches in a barometer, and at 62 degrees in a thermometer of Fah. Unfortunately, the expounders of this theory in England used only the generic term brass, and failed to define the specific gravity of the metal to be employed; the consequence of this omission is to leave room for an error of $\frac{1}{100000}$ in every attempt to reproduce or compare the results. This is the

minimum possible error; the maximum would be a fraction of the difference in specific gravity between the heaviest and lightest brass that can be cast.

The Wheel and Axle.—The wheel and axle may be considered as a perpetual lever, from the constant renewal of the points of suspension and resistance. The fulcrum is the centre of the axis, the longer arm is the radius of the wheel, and the shorter arm the radius of the axis. As the diameters of different circles bear the same proportion to each other that their respective circumferences do, the power is also to the weight as the diameter of the wheel is to the diameter of the axle. If one wheel move another of equal circumference, no power will be gained, as they will both move equally fast. But if one wheel move another of different diameter, whether larger or smaller, the velocities with which they move will be inversely as their diameters, circumferences, or number of teeth.

The Wedge. — The wedge is a double inclined plane, consequently its principles are the same. Hence, when two bodies are forced asunder, by means of the wedge, in a direction parallel to its head, multiply the resisting power by half the thickness of the head or back of the wedge, and divide the product by the length of one of its inclined sides; the quotient is the force equal to the resistance. The breadth of the back or head of a wedge being 3 inches, its inclined sides each 10 inches, required the power necessary to act upon the wedge so as to separate two substances whose resisting force is equal to 150 lbs. $\frac{150 \times 1{\cdot}5}{10} = 22{\cdot}5$ lbs.

Work is a term in mechanics of recent origin, but of great utility; it means a compound of force, pressure, and motion. Work is said to be performed when a pressure is exerted upon a body, and the body is thereby moved through space. The unit of pressure is a pound, the unit of space a foot; and work is measured by a foot-pound as a unit. Thus, if a pressure of so many pounds be exerted through a space of so many feet, the number of pounds is multiplied into the number of feet, and the product is the number of foot-pounds of work.

Metals and Alloys.

TABLE

OF MINERAL SUBSTANCES AND THEIR CHEMICAL EQUIVALENTS.

Names.	New Atomic Weights.	Old Atomic Weights.	Names.	New Atomic Weights.	Old Atomic Weights.
Aluminium	27·4	13·7	Manganese	55·0	27·5
Antimony	122·0	122·0	Mercury	200·0	100·0
Arsenic	75·0	75·0	Molybdenum	96·0	48·0
Barium	137·0	68·5	Nickel	59·0	29·5
Beryllium, or Glucinium	9·0	4·5	Niobium, or Columbium	94·0	94·0
Bismuth	209·0	209·0	Nitrogen	14·0	14·0
Boron	10·9	10·9	Osmium	199·0	99·5
Bromine	80·0	80·0	Oxygen	16·0	8·0
Cadmium	112·0	56·0	Palladium	106·5	53·25
Caesium	133·0	133·0	Phosphorus	31·0	31·0
Calcium	40·0	20·0	Platinum	197·4	98·7
Carbon	12·0	6·0	Potassium	39·11	39·11
Cerium	92·0	46·0	Rhodium	104·0	52·0
Chlorine	35·5	35·5	Rubidium	85·5	85·5
Chromium	52·5	26·25	Ruthenium	104·0	52·0
Cobalt	59·0	29·5	Selenium	79·5	39·75
Columbium, or Niobium	94·0	94·0	Silicon	28·0	14·0
			Silver	108·0	108·0
Copper	63·4	31·7	Sodium	23·0	23·0
Didymium	96·0	48·0	Strontium	87·5	43·8
Erbium	112·6	......	Sulphur	32·0	16·0
Fluorine	19·0	19·0	Tantalum	182·0	182·0
Gallium	......	......	Tellurium	129·0	64·5
Glucinium, or Beryllium	9·0	4·5	Terbium	148·5	74·2
			Thallium	204·0	204·0
Gold	196·0	98·0	Thorium	231·0	......
Hydrogen	2·0	1·0	Tin	118·0	59·0
Indium	114·0	57·0	Titanium	50·0	25·0
Iodine	127·0	127·0	Tungsten	184·0	92·0
Iridium	197·2	98·6	Uranium	120·0	60·0
Iron	56·0	28·0	Vanadium	51·0	51·0
Lanthamum	139·0	69·5	Yttrium	92·5	46·25
Lead	207·0	103·5	Zinc	65·0	32·5
Lithium	7·0	7·0	Zirconium	89·5	44·75
Magnesium	24·3	12·15			

Iron is the most important of all the metals known to man, as well as the most useful. It has been one of the principal agents

in the civilization of the human race, and is at the present day more extensively employed in the mechanical arts than any other metal. It is found in different conditions, but always in the state of oxides, or as iron ore, that is, a sort of rusty metallic state. The most common kind — the hematite or blood-stone — may be described as iron-rust solidified, or rendered concrete by water. After being taken from the ground in the condition of ore, it is placed in a blast-furnace and smelted, after which it is rendered fibrous and ductile by puddling. Spiegel iron or specular cast-iron is, as its name implies, largely crystalline, presenting bright, mirror-like, cleavage planes.

Wrought-iron varies in specific gravity from 7·8 to 7·6; taking the mean at 7·7, a cubic foot will weigh 479·8721664 lbs., or nearly 480 lbs. Cast-iron varies in specific gravity from 7·5 to 6·9, the average being 7·2.

	Wrought-Iron, Lbs.	Cast-Iron, Lbs.
A cubic foot...........	479·872	439·800
A cylindrical foot.....	376·891	344·407
A spherical foot.......	251·261	230·279
A cubic inch...........	0·2777	0·2845
A cylindrical inch....	0·2181	0·1999
A spherical inch......	0·1454	0·1333

Cast-iron is composed of about 91 per cent. of iron, 5 of carbon, 2 of silicon, and 2 parts of sulphur phosphorus, and other impurities. It also contains manganese, nickel, cobalt, chromium, vanadium, titanium, and tungsten, in minute quantities. The parts of steam-engines generally made of wrought-iron are the link, eccentric-rods and straps, valve- and piston-rods, connecting-rods, air-pump levers, cross-heads for pumps, arms, etc.

Rust. — The red powder that falls from iron which has long been subjected to the action of moisture, is the oxide of the metal, and is termed rust.

Steel is one of the chemical modifications of iron, a combination of iron and carbon. It is composed of 98·6 of iron and 1·4

of carbon. The steel containing the least carbon is the softest, and that containing the most is the hardest.

Cast-iron, wrought-iron, and steel can be distinguished from each other by the difference in the grain — wrought-iron being finer in the grain than cast, and steel finer than wrought; cast-iron being short and brittle, wrought-iron fibrous, and steel void of fibre.

Steel and cast-iron are fusible; wrought-iron is malleable, ductile, tough, fibrous, and possesses the quality of welding; steel, also, is capable of being welded. From this it will be seen, that steel possesses properties in common with both wrought- and cast-iron. Malleable iron is composed of 99·5 per cent. of iron, 0·035 of carbon, 0·076 of silicon, and the rest is sulphur and phosphorus. Its principal value consists in its property of resisting the chemical action of salt water or steam.

TABLE

SHOWING THE HEAT-CONDUCTING PROPERTIES OF DIFFERENT METALS.

	Conductive Property for Transmission of Heat.
Copper	1000.
Brass	468.
Wrought-Iron	336.
Cast-Iron	311.

From the above, it is evident that copper possesses the highest conducting properties.

TABLE

SHOWING THE TENACITY OR TENSILE STRENGTH OF DIFFERENT METALS.

	Tenacity in Lbs. per Square Inch.
Copper (cast)	19,000.
Brass (cast)	18,000.
Gun-Metal (cast)	36,000.
Iron, Wrought	51,000 to 61,000.
Iron, Cast	20,000.

Brass or gun-metal is used for main-bearings of marine-engines and propeller-shafts, link-blocks, air-pump buckets, head- and foot-valves, stern-tube bushes, propellers, and steam- and water-cocks. White metal is frequently used as a lining for main propeller-shaft and tunnel-bearings. Its chief value consists in its anti-friction and lubricating properties, while its disadvantages are that, if it becomes overheated, it will melt and run out of the bearing. Muntz metal is used for surface-condenser tubes, air- and circulating-pump rods, and surface-condenser tube-plates. It is malleable, has a high tensile strength, is very durable, and not liable to corrosion.

TABLE

SHOWING THE PROPORTION OF CARBON IN THE VARIOUS GRADES OF IRON AND STEEL.

Iron semi-steelified contains	1–150	of Carbon.
Soft steel capable of welding	1–120	"
Cast steel for common purposes	1–100	"
Cast steel requiring more hardness	1–90	"
Steel capable of standing a few blows, but quite unfit for drawing	1–50	"
First approach to a steely, granulated fracture	1–40	"
White cast-iron	1–25	"
Mottled cast-iron	1–20	"
Carbonated cast-iron	1–15	"
Super-carbonated crude iron	1–12	"

Copper is softer and more ductile than iron, is easily melted, and, when cast, is almost always free from blisters and sound. Its chief drawback is its cost and great weight, which are nearly double that of iron. Its superior conducting power is to some extent offset by the greater thickness required for strength.

Sulphur is less influenced by changes of temperature than any known mineral. It has a strong affinity for iron, and, as there is a great deal of it in bituminous coal, the sulphuretted hydrogen gas, disengaged from the fuel, attacks and soon destroys the metal.

Babbitt's Metal. — Its composition is as follows: Four pounds of copper, eight pounds of regulus of antimony, and eighty-eight pounds of tin. The copper is first melted; the tin and the regulus of antimony are then added. After the metals have been fused a short time, and brought to a dull red heat, it is fit for use.

Another durable alloy for the journal-boxes of steam-engines, is copper, 84; zinc, 8; tin, 2; lead, 4; and iron, 5 parts.

Bronze Alloy. — Copper, 80; tin, 18; zinc, 2. If, after casting, and while still red hot, cold water is poured over it, it becomes harder, and finer in grain, and tougher, as the tin, instead of separating, as happens, when the bronze cools slowly, remains mixed, and the alloy retains its compactness.

Alloys and Compositions.

	Copper.	Zinc.	Tin.	Antimony.	Lead.	Nickel.	Bismuth.
Brass for locomotive bearings.....	50·	2·5	5·	....	5		
Brass for glands.....................	65·	0·5	8·				
Brass engine bearings..............	50·	1·8	6·5				
Yellow brass for turning...........	40·	20·					
Brass richer............................	50·	10·					
Box metal...............................	80·	10·					
Red brass...............................	70·	10·	....	....	5		
Flanges to stand brazing...........	64·	2·	....	....	2		
Tough brass engine work..........	100·	15·	15·				
Tough brass for heavy bearings..	160·	5·	25·				
Muntz metal............................	90·	60·					
White metal............................	11·	11·	42·6	85·2			
White metal, hard....................	104·7	38·7	6·6				
Bronze red...............................	130·5	19·5					
Bronze yellow..........................	100·8	46·8	2·4				
Gun metal for bearings.............	90·3	9·67	0·3				
Bell metal for large bells..........	80·	....	20·				
Britannia metal.......................	1·	2·	81·	16·			
Brass for sheets......................	84·7	15·3					
Nickel-silver, English...............	60·	17·8	....	....	.....	22·2	
Nickel-silver, Parisian..............	66·	13·6	....	....	.....	19·3	
German silver.........................	50·	25·	....	....	.....	25·	

Solder.

Silver solder is generally composed of 4 parts silver and 2 parts yellow brass. Pure copper, in thin strips, is generally used for soldering-irons. Plumbers' solder is composed of 2 parts tin and 4 parts lead. This solder melts at about 450° Fah. Tinsmiths' solder is composed of 4 parts tin and 2 parts lead. This solder melts at about 350° Fah. Bismuth solder is composed of 7 parts bismuth, 5 parts lead, and 3 parts tin. This solder melts at about 225° Fah. All tin and lead solders become more fusible the more tin they contain. Thus, 1 part tin and 10 parts lead melt at about 550° Fah.; while 6 parts tin and 1 part lead melt at about 375° Fah. All the tin, lead, and bismuth solders become more fusible the more lead and bismuth they contain.

TABLE

SHOWING THE AVERAGE CRUSHING LOAD OF DIFFERENT MATERIALS, OR THE WEIGHT UNDER WHICH THEY WILL CRUMBLE.

	Lbs. per Sq. Inch.		Lbs. per Sq. Inch.
Alder	6,900	Walnut	6,000
Ash	8,600	Willow	2,900
Beech	7,600	Cast-iron, Am.	174,803
Cedar	5,700	Low moor, Eng	62,450
Elm	10,000	Wrought-iron	38,000
Fir — Spruce	6,500	Steel, cast	225,000
Hickory (white)	8,925	" tempered	337,800
Hornbeam	4,500	Copper, cast	117,000
Larch	3,200	Brass, "	164,800
Locust	9,113	Tin, "	15,500
Maple	8,150	Lead	7,730
Oak	4,200	Hard brick	2,000
" English	6,500	Crown glass	31,000
Pine (pitch)	6,800	Granite, Eng.	10,360
" Am. yellow	5,300	Portland cement	15,000
Poplar	5,100	Freestone, Conn.	3,522
Plum	3,700	Marble, Am.	18,061
Sycamore	7,000	Roman cement	342
Teak	12,000		

TABLE

SHOWING THE TENSILE STRENGTH, OR THE STRAIN THAT WILL PULL DIFFERENT METALS ASUNDER ON A STRAIGHT PULL.

	Lbs. per Sq. Inch.		Lbs. per Sq. Inch.
Antimony	1,000	Steel plates—Hussey, Wells & Co.—American	94,450
Bismuth	3,200	—— Bessemer—American	98,600
Brass—cast	18,000	Bessemer steel—tool	112,000
Copper—cast	19,000	Steel, bar—Black Diamond—American	120,700
Gun-metal, copper, and tin	96,000	—— tempered	214,400
Iron—cast	17,900	Chrome steel—American	180,000
Wrought-iron—bar	57,500	Silver—cast	41,000
—— good	60,000	Tin—block	4,600
—— superior	70,000	Zinc—cast	2,800
—— best American	76,160	" sheet	16,000
—— low moor	60,000	" wire	22,000
—— boiler-plate	45,000		
—— rivet—English	65,000		
Steel plates—English	78,000		

TABLE

SHOWING THE TENSILE STRENGTH OF DIFFERENT KINDS OF WOOD.

	Lbs. per Sq. Inch.		Lbs. per Sq. Inch.
Alder	14,000	Hickory	11,000
Ash	16,000	Lignum-Vitæ	11,000
Birch	15,000	Larch	7,000
Baywood	12,000	Locust	18,000
Beech	11,500	Maple	10,000
Bamboo	6,000	Mahogany	8,000
Boxwood	20,000	Oak	10,000
Cedar	7,000	Pear	10,000
Chestnut	13,000	Pine	10,000
Cypress	6,000	Poplar	7,000
Elder	10,000	Sycamore	12,000
Elm	6,000	Teak	15,000
Fir or Spruce	10,000	Walnut	8,000
Hazel	18,000	Yew	8,000
Holly	16,000		

Black Finish for Brass.—Make a strong solution of nitrate of silver in one dish and nitrate of copper in another. Mix the two together and plunge the brass into it. Now heat the brass evenly until the required degree of dead blackness is obtained. This is the method used by French instrument-makers to produce the beautiful dead-black color so much admired in optical instruments.

Lacquer for Brass Castings.—Take of shellac, 6 oz.; amber of copal, ground, 2 oz.; dragon's blood, 40 grains; extract of red sandal-wood, 30 grains; oriental saffron, 36 grains; pounded glass, 4 oz.; very pure alcohol, 44 oz. To apply to brass, expose to a gentle heat and dip them in.

Solder.—The following solder will braze steel or iron, and may be found very useful in case of a valve-stem or other light portion of an engine or machine breaking at a time when it is important that the engine or machine should continue work: Silver, 19 parts; copper, 1 part; brass, 2 parts.

Fusible Metal, consisting of 8 parts of bismuth, 5 of lead, and 3 of tin. It melts at the heat of boiling water, or 212° Fah. By the addition of a very little mercury, it becomes still more fusible, and is used for certain anatomical injections and for the filling of carious teeth.

Rule *for finding the approximate weight of iron castings from patterns.*—Multiply the weight of the pattern by the figures corresponding to the material in the table. Very accurate results cannot be expected, as the specific gravity of wood as well as of iron varies.

Pine wood	14·0
Oak "	9·0
Beech "	9·7
Linden "	13·4
Birch "	10·6
Alder "	12·6
Pear-tree wood	10·0

TABLE

SHOWING THE WEIGHT OF CASTINGS BY WEIGHT OF THE PATTERNS.

Multiply the weight of the pattern by the multiplier opposite each material.

White Pine	× 16	Cast-iron.
"	× 17·1	Wrought-iron.
"	× 17·3	Steel.
"	× 18	Copper.
"	× 25	Lead.

TABLE

SHOWING THE SHRINKAGE OF CASTINGS OF DIFFERENT METALS.

Cast-iron, $\frac{1}{8}$ inch per lineal foot.
Brass, $\frac{3}{16}$ " "
Lead, $\frac{1}{8}$ " "
Tin, $\frac{1}{12}$ inch per lineal foot.
Zinc, $\frac{5}{16}$ " "

TABLE

SHOWING THE WEIGHT AND BULK OF DIFFERENT SUBSTANCES IN CUBIC FEET, POUNDS AND TONS.

Names of Substances.	Pounds in One Cubic Foot.	Cubic Feet in one gross Ton.	Names of Substances.	Pounds in One Cubic Foot.	Cubic Feet in one gross Ton.
Cast-iron	450·5	4·97	Oak, white	45·2	49·5
Wrought-iron	486·6	4·60	Clay	101·3	22·1
Steel	489·8	4·57	Concrete, ordinary	115·0	19·5
Copper	555·0	4·03	Brick	100·0	22·4
Lead	707·0	3·16	Plaster, Paris	105·0	21·3
Brass	537·7	4·16	Sand	94·5	23·7
Tin	456·0	4·91	Granite	139·0	16·1
Pine, white	29·56	75·6	Earth, loose	78·6	28·5
" yellow	33·81	66·2	Water, salt (sea)	64·3	34·8
Mahogany	66·4	33·8	Water, fresh	62·5	35·9
Marble, common	141·0	15·9	Ice	58·08	38·56
Millstone	130·0	17·2	Gold	1013·0	2·21
Oak, live	70·0	32·0	Silver	551·0	4·07

TABLE

SHOWING THE WEIGHT OF DIFFERENT METALS PER CUBIC FOOT.

	Lbs.		Lbs.
Brass	525	Lead, cast	710
Copper	550	Silver	655
Gold	1,210	Steel	490
Iron, cast	450	Tin, cast	456
Iron, wrought	485	Zinc	450

TABLE

SHOWING THE ACTUAL EXTENSION OF WROUGHT-IRON AT VARIOUS TEMPERATURES.

Deg. of Fah.	Length.	
32°	1·	
212	1·0011356	
392	1·0025757	Surface becomes straw-colored, deep-yellow, crimson, violet, purple, deep-blue, bright-purple.
672	1·0043253	
752	1·0063894	
932	1·0087730	Surface becomes dull, and then bright-red.
1,112	1·0114811	
1,652	1·0216024	Bright-red, yellow, welding heat, white heat.
2,192	1·0348242	
2,732	1·0512815	
2,912	Cohesion destroyed. Fusion perfect.	

Linear Expansion of Wrought-Iron. — The linear expansion which a bar of wrought-iron undergoes, according to Daniell's pyrometer, when heated from the freezing- to the boiling-point, or from 32° to 212° Fah., is about $\frac{1}{880}$ of its length; at higher temperatures the elongation becomes more rapid. Thus, it will be seen how sensible a change takes place when iron undergoes a variation of temperature. A bar of iron 10 feet long, subject to an ordinary change of temperature of from 32° to 180° Fah., will elongate more than $\frac{1}{8}$ of an inch, or sufficient to cause fracture in stone-work, strip the thread of a screw, or endanger a bridge,

floor, roof, or truss, or even push out a wall if brought in contact with it.

The expansion of volume and surface of wrought-iron is calculated by taking the linear expansion as unity; then, following the geometrical law, the superficial expansion is twice the linear, and the cubical expansion is three times the linear.

Wrought-iron will bear on a square inch, without permanent alteration, 17,800 pounds, and an extension in length of $\frac{1}{1400}$. Cohesive force is diminished $\frac{1}{3000}$ by an increase of one degree of heat.

Compared with cast-iron, its strength is 1·12 times, its extensibility 0·86 times, and its stiffness 1·3 times.

Cast-iron expands $\frac{1}{162000}$ of its length for one degree of heat; the greatest change in the shade, in this climate, is $\frac{1}{1170}$ of its length; exposed to the sun's rays, $\frac{1}{1000}$.

Cast-iron shrinks, in cooling, from $\frac{1}{85}$ to $\frac{1}{98}$ of its length.

Cast-iron is crushed by a force of 93,000 pounds upon a square inch, and will bear, without permanent alteration, 15,300 pounds upon a square inch.

To find the surface dilatation of any particular article, double its linear dilatation; and to find the dilatation in volume, triple it. To find the elongation in linear inches, per linear foot, of any particular article, multiply its respective linear dilatation, as given in the table, by 12.

TABLE

SHOWING THE LINEAR DILATATION OF SOLIDS BY HEAT.

Length which a Bar Heated at 212° has greater than when at the Temperature of 32°.

Brass, cast	·0018671
Copper	·0017674
Gold	·0014880
Iron, cast	·0011111
Iron, wrought	·0012575
Silver	·0020205
Steel	·0011898

TABLE

DEDUCED FROM EXPERIMENTS ON IRON PLATES FOR STEAM-BOILERS, BY THE FRANKLIN INSTITUTE, PHILADA.

Iron boiler-plate was found to increase in tenacity, as its temperature was raised, until it reached a temperature of 550° above the freezing-point, at which point its tenacity began to diminish.

At 32° to 80°	tenacity is	56,000 lbs.,	or	$\frac{1}{7}$ below its maximum.
" 570°	"	" 66,000	"	the maximum.
" 720°	"	" 55,000	"	the same nearly as at 30°.
" 1050°	"	" 32,000	"	nearly $\frac{1}{2}$ the maximum.
" 1240°	"	" 22,000	"	nearly $\frac{1}{3}$ the maximum.
" 1317°	"	" 9,000	"	nearly $\frac{1}{7}$ the maximum.

It will be seen by the above table that if a boiler should become overheated by the accumulation of scale on some of its parts, or an insufficiency of water, the iron would soon become reduced to less than one-half its strength.

TABLE

SHOWING THE STRENGTH OF COPPER BOILER PLATES AT DIFFERENT TEMPERATURES, DEDUCED FROM EXPERIMENTS BY THE FRANKLIN INSTITUTE OF PHILA. THE STANDARD STRENGTH AT 32° BEING 32,800 LBS. PER SQUARE INCH.

	Temperature above 32°.	Diminution of Strength.		Temperature above 32°.	Diminution of Strength.
1	90°	0·0175	9	660°	0·3425
2	180	0·0540	10	769	0·4398
3	270	0·0926	11	812	0·4944
4	360	0·1513	12	880	0·5581
5	456	0·2046	13	989	0·6691
6	460	0·2133	14	1000	0·6741
7	513	0·2446	15	1200	0·8861
8	532	0·2558	16	1300	1·000

It will be seen from the above table, that, in being heated from the freezing-point to the boiling-point of water, copper loses 5 per cent. of its strength; at 550° it loses about one-quarter of its strength; and at 1332° loses all its tenacity.

TABLE

SHOWING THE WEIGHT OF CAST-IRON BALLS FROM 3 TO 13 INCHES IN DIAMETER.

Diam. in Inches..	3	3¼	3½	3¾	4	4¼	4½	4¾	5	5¼	5½	5¾	6	6¼	6½	6¾	7	7¼	7½	7¾
Wt. in Pounds...	3¾	4¾	5¾	7¼	8¾	10½	12½	14¾	17	20	23	26	29¾	33½	37¾	42¼	47¼	52½	58	64

Diam. in Inches...	8	8¼	8½	8¾	9	9¼	9½	9¾	10	10¼	10½	10¾	11	11½	12	13
Wt. in Pounds.....	70½	77¼	84¼	92½	100¼	109	118	127½	137¾	148¼	159½	171	183¼	209½	238	302

To find the weight of any cast-iron ball, cube the diameter, and multiply by the decimal ·5236; the product will be the number of cubic inches in the ball, which, if multiplied by the decimal ·2607, will give the weight of the ball in pounds.

WEIGHT OF CAST-IRON PLATES PER SUPERFICIAL FOOT AS PER THICKNESS.

Thickness in Inches..........	⅛		¼		⅜		½		⅝		¾		⅞		1	
	lbs.	oz.	lbs.	oz.	lbs.	oz.	lbs.	oz.	lbs.	oz.	lbs.	oz.	lbs.	oz.	lbs.	oz.
Weight........................	4	13	9	10	14	8	19	5	24	2	29	0	33	13	38	10

TABLE

SHOWING THE WEIGHT OF ROUND-IRON FROM ½ AN INCH TO 6 INCHES IN DIAMETER, 1 FOOT LONG.

For Calculating the Weight of Shafting, etc.

Diameter in Inches........	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2	2⅛
Weight in Pounds.........	¾	1	1½	2	2¾	3½	4¼	5	6	7	8	9½	10¼	12

Diameter in Inches.......	2¼	2⅜	2½	2⅝	2¾	2⅞	3	3⅛	3¼	3⅜	3½	3⅝	3¾	3⅞
Weight in Pounds.........	13½	15	16¾	18¼	20	22	24	26	28	30¼	32½	35	37¼	40

Diameter in Inches.......	4	4⅛	4¼	4⅜	4½	4⅝	4¾	4⅞	5	5¼	5½	5¾	6
Weight in Pounds.........	42½	45¼	48	50¾	53¾	56¾	60	63	66¾	73¼	80¼	87¾	95½

TABLE

SHOWING THE WEIGHT OF BOILER-PLATES 1 FOOT SQUARE AND FROM $\frac{1}{16}$TH TO AN INCH THICK.

Thick. in Ins..	Wt. in Lbs. per ft. sq.
$\frac{1}{16}$	$2\frac{1}{2}$
$\frac{1}{8}$	5
$\frac{3}{16}$	$7\frac{1}{2}$
$\frac{1}{4}$	10
$\frac{5}{16}$	$12\frac{1}{2}$
$\frac{3}{8}$	15
$\frac{7}{16}$	$17\frac{1}{2}$
$\frac{1}{2}$	20
$\frac{9}{16}$	$22\frac{1}{2}$
$\frac{5}{8}$	25
$\frac{11}{16}$	$27\frac{1}{2}$
$\frac{3}{4}$	30
$\frac{13}{16}$	$32\frac{1}{2}$
$\frac{7}{8}$	35
$\frac{15}{16}$	$37\frac{1}{2}$
1	40

TABLE

SHOWING THE WEIGHT OF SQUARE BAR-IRON, FROM $\frac{1}{2}$ AN INCH TO SIX INCHES SQUARE, 1 FOOT LONG.

Square........	Wt. in Lbs....
$\frac{1}{2}$	$\frac{8}{10}$
$\frac{5}{8}$	$1\frac{1}{4}$
$\frac{3}{4}$	2
$\frac{7}{8}$	$2\frac{1}{2}$
1	$3\frac{1}{2}$
$1\frac{1}{8}$	$4\frac{1}{4}$
$1\frac{1}{4}$	$5\frac{1}{4}$
$1\frac{3}{8}$	$6\frac{1}{2}$
$1\frac{1}{2}$	$7\frac{1}{2}$
$1\frac{5}{8}$	9
$1\frac{3}{4}$	$10\frac{1}{2}$
$1\frac{7}{8}$	12
2	$13\frac{1}{2}$
$2\frac{1}{8}$	$15\frac{1}{4}$
$2\frac{1}{4}$	17
$2\frac{3}{8}$	19
$2\frac{5}{8}$	22
$2\frac{1}{2}$	$23\frac{1}{4}$
$2\frac{3}{4}$	$25\frac{1}{2}$
$2\frac{7}{8}$	28
3	$30\frac{1}{2}$
$3\frac{1}{8}$	33
$3\frac{1}{4}$	$35\frac{3}{4}$
$3\frac{3}{8}$	$38\frac{1}{2}$
$3\frac{1}{2}$	$41\frac{1}{2}$
$3\frac{5}{8}$	$44\frac{1}{2}$
$3\frac{3}{4}$	$47\frac{1}{2}$
$3\frac{7}{8}$	$50\frac{3}{4}$
4	54
$4\frac{1}{8}$	$57\frac{1}{2}$
$4\frac{1}{4}$	61
$4\frac{3}{8}$	$64\frac{3}{4}$
$4\frac{1}{2}$	$68\frac{1}{2}$
$4\frac{5}{8}$	$72\frac{1}{2}$
$4\frac{3}{4}$	$76\frac{1}{4}$
$4\frac{7}{8}$	$80\frac{1}{4}$
5	$84\frac{1}{2}$
$5\frac{1}{4}$	$93\frac{1}{4}$
$5\frac{1}{2}$	$102\frac{1}{4}$
$5\frac{3}{4}$	$111\frac{3}{4}$
6	$121\frac{3}{4}$

TABLE

SHOWING THE WEIGHT OF CAST-IRON PIPES, 1 FOOT IN LENGTH, FROM ¼ INCH TO 1¼ INCHES THICK AND FROM 3 INCHES TO 24 INCHES DIAMETER.

Diam. of Bore in Inches.	Thickness in Inches.								
	¼ Lbs.	⅜ Lbs.	½ Lbs.	⅝ Lbs.	¾ Lbs.	⅞ Lbs.	1 Lbs.	1⅛ Lbs.	1½ Lbs.
3	8½	12½	17¼	22¼	27½	......	......	......	......
3½	9¼	14¼	19½	25¼	31¼	......	......	......	......
4	10	16¾	22	28½	35	......	......	......	......
4½	11¾	18	24½	31½	38¾	......	......	......	......
5	13	19¾	27	34½	42¼	50½	59	......	......
5½	15	21½	29½	37½	46	54¾	63¾	......	......
6	......	23½	32	40¾	49¾	59	68¾	78¾	88¾
6½	......	25¼	34½	43¾	53½	63½	73½	84¼	95
7	......	27¼	36¾	46¾	56¾	67¾	78½	89¾	101¼
7½	......	29	39	50	60¾	72	83½	95¼	107½
8	......	30¾	41¾	53	64½	76¼	88½	100¾	113½
8½	......	33	44½	56¼	68¼	80¾	93½	106½	120
9	......	34½	46½	59	71¾	84¾	98½	111¾	125¾
9½	......	36¼	49	62	75½	89	103	117½	132
10	......	38¼	51½	65¼	79¼	93½	108	122¾	138
10½	......	......	54	68¼	82¾	97¾	112¾	128½	144¼
11	......	......	56½	71¼	86½	102	117¾	134	150¼
11½	......	......	59	76¼	90	106¼	122¾	139½	156½
12	......	......	61¼	77½	93½	110½	127½	145	162½
13	......	......	......	82¾	101¼	118¼	137½	154	173½
14	......	......	......	89¼	108¼	126½	146¼	165¼	185¼
15	......	......	......	95¼	115¾	135¼	156¼	176¼	198
16	......	......	......	......	123¼	143	166	187½	211¼
17	......	......	......	......	130¼	152½	178½	198¼	223½
18	......	......	......	......	137	161¼	185¼	209	235¼
19	......	......	......	......	......	169¼	195¾	222¼	247
20	......	......	......	......	......	178	205¼	233¼	259
21	......	......	......	......	......	......	214	243½	273¼
22	......	......	......	......	......	......	223½	244¾	285¼
23	......	......	......	......	......	......	233½	265½	298¼
24	......	......	......	......	......	......	245¼	277½	310½

TABLE

SHOWING THE STANDARD WEIGHTS OF CAST-IRON WATER-PIPE.

3 inch, 15	lbs.	per foot	= 180	lbs.	per	length	of 12	feet.
4 inch, 22	"	"	= 264	"	"	"	"	
6 inch, 33	"	"	= 400	"	"	"	"	
8 inch, 42	"	"	= 500	"	"	"	"	
10 inch, 60	"	"	= 720	"	"	"	"	
12 inch, 75	"	"	= 900	"	"	"	"	

TABLE

SHOWING THE STANDARD WEIGHTS OF CAST-IRON GAS-PIPE.

3 inch, 12½	lbs.	per foot	= 150	lbs.	per	length	of 12	feet.
4 inch, 17	"	"	= 204	"	"	"	"	
6 inch, 30	"	"	= 360	"	"	"	"	
8 inch, 40	"	"	= 480	"	"	"	"	
10 inch, 50	"	"	= 600	"	"	"	"	
12 inch, 70	"	"	= 840	"	"	"	"	

TABLE

SHOWING THE TENSILE STRENGTH OF VARIOUS QUALITIES OF AMERICAN CAST-IRON.

	Breaking weight of a square inch bar.
Common pig-iron,	15,000
Good common castings,	20,000
Cast-iron "	20,834
" "	19,200
" "	27,700
Gun-heads, specimen from,	24,000
" " "	39,500
Greenwood cast-iron,	21,300
" " (after third melting,)	45,970
Mean of American cast-iron,	31,829
Gun-metal, mean,	37,232

English Cast-Iron.

	Breaking weight of a square inch bar.
Low Moor,	14,076
Clyde, No. 1,	16,125
Clyde, No. 3,	23,468
Calder, No. 1,	13,735
Stirling, mean,	25,764
Mean of English,	19,484
Stirling, toughened iron,	28,000
Carron No. 2, cold-blast,	16,683
" " 2, hot-blast,	13,505
" " 3, cold-blast,	13,200
" " 3, hot-blast,	17,755
Davon, No. 3, hot-blast,	21,907
Buffery, No. 1, cold-blast,	17,466
" " 1, hot-blast,	13,437
Cold-Talon (North Wales), No. 2, cold-blast,	18,855
" " " " 2, hot-blast,	16,676

TABLE

SHOWING THE TENSILE STRENGTH OF VARIOUS QUALITIES OF AMERICAN WROUGHT-IRON.

	Breaking weight of a square inch bar.
From Salisbury, Conn.,	66,000
" Pittsfield, Mass.,	57,000
" Bellefonte, Pa.,	58,000
" Maramec, Mo.,	43,000
" " "	53,000
" Centre County, Pa.,	58,400
" Lancaster County, Pa.,	58,061
" Carp River, Lake Superior,	89,582
" Mountain, Mo., Charcoal bloom,	90,000
American hammered,	53,900
Chain-iron,	43,000
Rivets,	53,300

	Breaking weight of a square inch bar.
Bolts,	52,250
Boiler-plates,	50,000
Average boiler-plates,	55,000
" joints, double-riveted,	35,000
" " single "	28,600
Chrome steel, highest strength,	198,910
" " lowest "	163,760
" " average "	180,000
Homogeneous metal,	105,732
" " 2d quality,	81,663
Bessemer steel,	148,324
" "	154,825
" "	157,881

TABLE

SHOWING THE RESULTS OF EXPERIMENTS MADE ON DIFFERENT BRANDS OF BOILER-IRON AT THE STEVENS INSTITUTE OF TECHNOLOGY, HOBOKEN, N. J.

Thirty-three experiments were made upon the iron taken from the exploded steam-boiler of the ferry-boat "Westfield." The following were the results:

	Lbs. per sq. in.
Average breaking weight,	41,653
16 experiments made upon high grades of American boiler-plate.	
Average breaking weight,	54,123
15 experiments made upon high grades of American flange-iron.	
Average breaking weight,	42,144
6 experiments made upon English Bessemer steel.	
Average breaking weight,	82,621
5 experiments made upon English Low Moor boiler-plate.	
Average breaking weight,	58,984

Lbs. per sq. in

6 experiments made upon samples of tank-iron taken from different manufacturers.

Average breaking weight No. 1, ... 43,831

" " " No. 2, ... 42,011

" " " No. 3, ... 41,249

2 experiments made on iron taken from the exploded steam-boiler of the "Red Jacket."

Average breaking weight, ... 49,000

It will be noticed that the above experiments reveal a great variation in the strength of boiler-plate of different grades.

TABLE

GIVING THE PROPORTIONS OF THE UNITED STATES OR SELLERS' STANDARD THREADS FOR SCREWS, NUTS, AND BOLTS.

Outside Diameter of Screw in Inches.	Number of Threads per Inch.	Diameter of Screw at the Root of the Thread in Decimals of an Inch.	Width of Top and Bottom of Thread in Decimals of an Inch.	Outside Diameter of Screw in Inches.	Number of Threads per Inch.	Diameter of Screw at the Root of the Thread in Decimals of an Inch.	Width of Top and Bottom of Thread in Decimals of an Inch.
$\frac{1}{4}$	20	·185	·0062	**2**	$4\frac{1}{2}$	1·712	·0277
$\frac{5}{16}$	18	·240	·0074	$2\frac{1}{4}$	$4\frac{1}{2}$	1·962	·0277
$\frac{3}{8}$	16	·294	·0078	$2\frac{1}{2}$	4	2·176	·0312
$\frac{7}{16}$	14	·344	·0089	$2\frac{3}{4}$	4	2·426	·0312
$\frac{1}{2}$	13	·400	·0096	**3**	$3\frac{1}{2}$	2·629	·0357
$\frac{9}{16}$	12	·454	·0104	$3\frac{1}{4}$	$3\frac{1}{2}$	2·879	·0357
$\frac{5}{8}$	11	·507	·0113	$3\frac{1}{2}$	$3\frac{1}{4}$	3·100	·0384
$\frac{3}{4}$	10	·620	·0125	$3\frac{3}{4}$	3	3·317	·0413
$\frac{7}{8}$	9	·731	·0138	**4**	3	3·567	·0413
1	8	·837	·0156	$4\frac{1}{4}$	$2\frac{7}{8}$	3·798	·0435
$1\frac{1}{8}$	7	·940	·0178	$4\frac{1}{2}$	$2\frac{3}{4}$	4·028	·0454
$1\frac{1}{4}$	7	1·065	·0178	$4\frac{3}{4}$	$2\frac{5}{8}$	4·256	·0476
$1\frac{3}{8}$	6	1·160	·0208	**5**	$2\frac{1}{2}$	4·480	·0500
$1\frac{1}{2}$	6	1·284	·0208	$5\frac{1}{4}$	$2\frac{1}{2}$	4·730	·0500
$1\frac{5}{8}$	$5\frac{1}{2}$	1·389	·0227	$5\frac{1}{2}$	$2\frac{3}{8}$	4·953	·0526
$1\frac{3}{4}$	5	1·491	·0250	$5\frac{3}{4}$	$2\frac{3}{8}$	5·203	·0526
$1\frac{7}{8}$	5	1·616	·0250	**6**	$2\frac{1}{4}$	5·423	·0555

The "pitch" of a thread is the distance which it travels lengthways for one revolution of the screw.

The thickness or depth of a nut, to give equal strength, must be equal to the outside diameter of the screw or bolt.

Speed, Power, Capacity, and Dress of Millstones.

Diameter of Millstone.		Revolutions per Minute.	Horse-Power.	Average Capacity per Hour of Grinding in Bushels.	Usual Dress.	Draught from Fore Edge of Furrow.
Ft.	In.					Inches.
2	6	200	$2\frac{1}{2}$	$2\frac{1}{2}$	7·3	$2\frac{1}{2}$
2	10	180	$2\frac{3}{4}$	$2\frac{3}{4}$	8·2	$2\frac{1}{2}$
3	0	170	3	3	9·3	$2\frac{1}{2}$
3	2	160	$3\frac{1}{4}$	$3\frac{1}{4}$	9·3	$2\frac{3}{4}$
3	4	150	$3\frac{1}{2}$	$3\frac{1}{2}$	10·3	3
3	6	140	$3\frac{3}{4}$	$3\frac{3}{4}$	10·3	3
3	8	130	$3\frac{7}{8}$	$3\frac{7}{8}$	10·3	3
3	10	125	$3\frac{7}{8}$	Nearly 4	11·3	3
4	0	120	4	4	10·4	3
4	2	115	$4\frac{1}{8}$	$4\frac{1}{4}$	10·4	3
4	4	110	$4\frac{1}{4}$	$4\frac{1}{2}$	11·4	$3\frac{1}{4}$
4	6	105	$4\frac{1}{2}$	5	12·4	$3\frac{1}{2}$
4	8	100	$4\frac{3}{4}$	6	12·4	$3\frac{3}{4}$
4	10	95	5	$6\frac{1}{2}$	12·4	4
5	0	90	6	7	12·4	$4\frac{1}{2}$

Speed of Circular-Saws.

About nine thousand feet per minute for the rim of a circular-saw to travel may be laid down as good speed: a saw twelve inches in diameter, 3 feet around the rim, 3,000 revolutions; 24 inches in diameter, or 6 feet around the rim, 1,500 revolutions; 3 feet in diameter, or 9 feet around the rim, 1,000 revolutions; 4 feet in diameter, or 12 feet around the rim, 750 revolutions; 5 feet in diameter, or 15 feet around the rim, 600 revolutions.

Rule *for finding the proper number of revolutions per minute of any sized saw.*—Divide 36,000 by the diameter of the saw in inches; the quotient will be the right number of revolutions.

Example.—36,000 divided by 60 equals 600, the number of revolutions a 60-inch saw should make.

TABLE

OF COEFFICIENTS OF FRICTIONS BETWEEN PLANE SURFACES.

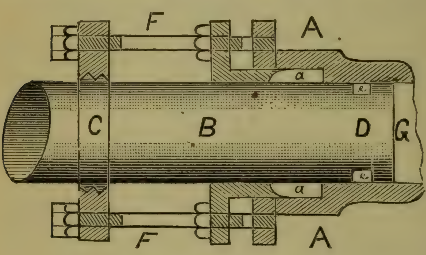
Sliding Surface.	Surface at Rest.	State of the Surfaces.		Coefficient of Friction.
Cast-iron.	Wrought-iron.	Fibres of both surfaces parallel to motion.	surfaces unctuous	0·143
			without lubricant	0·152
			surfaces unctuous	0·144
Cast-iron.	Cast-iron.	"	lubricated with tallow	0·100
			lubricated with lard	0·070
			lubricated with olive-oil	0·064
			lubricated with lard and plbg.	0·055
Wrought-iron.	Bronze.	Fibres parallel to motion.	without lubricant	0·072
			surfaces unctuous	0·060
			lubricated with tallow	0·103
			lubricated with lard	0·075
			lubricated with olive-oil	0·078
Bronze.	Wrought-iron.	"	without lubricant	0·161
			surfaces unctuous	0·166
			lubricated with tallow	0·081
			lubricated with lard and plbg.	0·089
			lubricated with olive-oil	0·072
Cast-iron.	Bronze.	"	without lubricant	0·147
			surfaces unctuous	0·132
			lubricated with tallow	0·103
			lubricated with lard	0·075
			lubricated with olive-oil	0·078
Bronze.	Cast-iron.	"	without lubricant	0·217
			surfaces unctuous	0·107
			lubricated with tallow	0·086
			lubricated with olive-oil	0·077
Bronze.	Bronze.	"	without lubricant	0·201
			surfaces unctuous	0·134
			lubricated with olive-oil	0·058
Brass.	Cast-iron.	"	without lubricant	0·189
			surfaces unctuous	0·115
			lubricated with tallow	0·072
			lubricated with lard	0·068
			lubricated with olive-oil	0·066
Steel.	"	"	without lubricant	0·202
			lubricated with tallow	0·105
			lubricated with lard	0·081
			lubricated with olive-oil	0·079
Steel.	Wrought-iron.	Fibres of iron parallel to motion.	lubricated with tallow	0·093
			lubricated with lard	0·076
Steel.	Bronze.		without lubricant	0·152
			lubricated with tallow	0·056
			lubricated with olive-oil	0·053
			lubricated with lard and plbg.	0·076

Non-Conducting Covering for Steam-Boilers and Pipes.

Make a thin paste by boiling flour and water, then stir in as much sawdust as it can hold together. After drying, it will adhere to iron when slightly warm, after which several coats may be applied in succession. It may be made water-proof by painting with coal-tar.

Or, mix thoroughly equal parts of fuller's-earth and finely-sifted coal-ashes; add to the pasty mass a small quantity of calves-hair. Before laying it on a pipe, add ⅛ its quantity of calcined gypsum, and lay on in thin layers. The pipe or cylinder should be warm when the mass is applied.

The annexed cut represents what is known to mechanics as an expansion-joint, such as is used on short, straight steam-pipes in contracted places, for the purpose of preventing them from breaking or throwing the machinery out of line by the force of expansion. *A* shows the socket, or stuffing-box; *B*, the tube or pipe; *C*, the guide employed for the purpose of keeping the pipe straight; *F F*, the studs by which the gland is adjusted, and the guide, *C*, retained in position; *a a* is the cavity which contains the fibrous packing for the purpose of preventing leakage; *e e*, the recess into which the head, *D*, is drawn when the pipe contracts; and *G* the female tube into which it is forced by expansion.



Steam-Joints.

Cement for steam-joints *and patching steam-boilers.* — Take any desired quantity of pure red lead, put it in an iron mortar, on a block or thick plate of iron. Put in a quantity of white lead ground in oil; knead them until you make a thick putty; then pound it; the more it is pounded the softer it will become. Roll

in red lead, and pound again; repeat this operation of adding red lead and pounding, until the mass becomes a stiff putty. In applying it to the flange or joint, it is well to put a thin grummet around the orifice of the pipe, to prevent the cement being forced inward to the pipe when the bolts are screwed up. When the flanges are not faced, make the above mass rather soft, and add cast-iron borings run through a fine sieve, when it will be found to resist either fire or water.

Or, powdered litharge, 2 parts; very fine sand, 2 parts; slaked quicklime, 1 part. Mix all together. So use. Mix the proper quantity with boiled linseed-oil, and apply quickly. It gets hard very soon.

Or, white lead ground in oil, 10 parts; black oxide of manganese, 3 parts; litharge, 1 part. Reduce to the proper consistency with boiled linseed-oil, and apply.

Or, red lead ground in oil, 6 parts; white lead, 3 parts; oxide of manganese, 2 parts; silicate of soda, 1 part; litharge, ½ part; all mixed and used as putty.

Or, take 10 pounds of ground litharge; 4 lbs. of ground Paris white; ½ pound of yellow ochre and ½ oz. of hemp; cut into lengths of ½ inch; mix all together with boiled linseed-oil to the consistency of a stiff putty. This cement resists fire and will set in water.

Cement for rust-joints. — Cast-iron borings or turnings, 19 lbs.; pulverized sal-ammoniac, 1 pound; flour of sulphur, ½ pound. It should be thoroughly mixed, and passed through a tolerably fine sieve. Sufficient water should be added to wet the mixture through. It should be prepared some hours before use. A small quantity of sludge from the trough of a grindstone will improve its quality.

All movable joints of the best description of land- and marine-engines are now faced on a lathe or planer, and then rendered perfectly steam-, air-, and water-tight by filing and scraping, so that all that is necessary, when put together, is to oil their surfaces.

For smooth surfaces that can be conveniently calked, sheet-copper, annealed by heating to a cherry red and then plunging it into cold water, makes a permanent joint.

Lead-wire makes a very cheap, clean, and permanent joint. Copper-wire also makes a good joint; but, when convenient, it is always best to plane or turn a groove in one of the surfaces to be brought into contact.

For uniform surfaces, gauze wire-cloth coated on either side with white or red lead paint makes a very durable joint, particularly where it is exposed to high temperatures.

For pumps or stand-pipes in the holds of vessels, canvas well saturated on both sides with white or red lead makes a very durable joint. Pasteboard painted on both sides with white or red lead paint is frequently used with good results.

How to make a good adhesive cement. — Mix pulverized gum-Arabic with its weight of finely-powdered calcined alum. When mixed with a small quantity of water, it forms a cement which unites wood, paper, porcelain, glass, and crockery very firmly. It must be kept dry in powder and moistened only as needed.

A cement for leather may be made by dissolving in a mixture of ten parts of bi-sulphide of carbon and one part of oil of turpentine enough gutta-percha to thicken the composition. The leather must be freed from grease, which may be done by placing a cloth between the leather and a hot iron. The pieces cemented must be pressed together until the cement is dry.

A cement for fastening leather to iron, china, or glass. — To one quart of glue dissolved in good cider vinegar add one ounce of good Venice turpentine. It should be allowed to simmer about half a day.

Cement for leather belting. — Of common glue and American isinglass, take equal parts; place in a kettle, and add sufficient water to cover the whole. Let them soak ten hours; then bring the mixture to the boiling point, and add pure tannin, until the whole becomes ropy, or appears like the white of eggs. Apply it warm. Buff the grain off the leather where it is to be cemented;

rub the joint surfaces solidly together, and let it dry for a few hours.

Cement for rubber belting. — Take 16 parts of gutta-percha or India-rubber; 2 parts common pitch; and 1 part linseed-oil. Melt together, and use hot. This cement will unite leather or rubber that has not been vulcanized.

Cement for brass and glass. — Boil 3 parts of resin with 1 part of caustic soda and 5 of water. Add five times its weight of plaster of Paris. It sets firmly in from half- to three-quarters of an hour. Zinc, white lead, or precipitated chalk may be substituted for plaster, but hardens more slowly.

Cement for stone or marble. — The best cement for mending marble, or any kind of stone, is made by mixing 20 parts of litharge and 1 part of freshly-burned lime in fine, dry powder. This is made into a putty by the addition of linseed-oil. It sets in a few hours, having the appearance of light stone.

Belting.

While the use of belts for the transmission of power is not an American invention, the numerous improvements made in this country have caused it to be known in Europe as the American system. In Europe the greater part of the power is transmitted by cog-wheels, but in this country 99 per cent. is transmitted by belting. The latter is used everywhere, from the sewing-machine to the 500 horse-power engine of the largest factory.

Belts can be run in any way, at any angle, of any length, and at any speed, and can be put up by any one of ordinary skill. They can be made of any flexible material — leather, rubber, gutta-percha, or cloth; yet, while so hardy and so popular, they have one fault — they are not positive. If the motion makes a certain number of revolutions, a portion of them is lost with every belt used. This is the only fault of the system. It is noiseless, yielding, and regular; but, unlike cog-wheels, it is not posi-

tive. The number of revolutions that are lost may, and do, vary continually by changes of the load or of the atmosphere.

Belts derive their power to transmit motion from the friction between the surface of the belt and the pulley, and from nothing else, and are governed by the same laws as in friction between flat surfaces. The friction increases regularly with the pressure. The great difference often observed in the friction of belts is due simply to their elasticity of surface; that is, the more elastic the surface the greater the friction.

In taking power from any source of motion, there are two points which control us; all the others we can control and modify to a certain extent. Ordinary belts will sustain safely a working tension of 45 lbs. per inch in width; the rule to determine the width of belt and size of pulley required to transmit a given horse-power is easily found. Since a horse-power is 33,000 pounds raised one foot high per minute, we must adjust the width and velocity of belts so as to effect the required result. Thus, if the belt moves with a velocity of 733 feet per minute, a belt five inches wide will transmit five horse-power, provided the effective tension is 45 lbs. per inch. If the velocity be increased to 1466 feet per minute, the same belt with the same tension will transmit ten horse-power. So that a five-inch belt, applied to a five-foot pulley making 120 revolutions per minute, would transmit ten horse-power when the effective tension is 225 pounds.

By taking the actual effective tension of the belt, and multiplying it by the actual velocity, we get what may be called the indicated horse-power of the belt, which corresponds to the indicated horse-power of the engine. And, finally, by measuring the actual power transmitted — which may be done by means of a dynamometer — we can get the actual power transmitted. Rules based upon the amount of belt surface in contact with the pulley, and on similar data, cannot be made to give reliable results. For practical purposes, velocity and power to resist tension are the only available elements of the calculation. Actual tension, adhesion, friction, etc., can all be varied at will, and consequently form

no certain dependence for the calculations of the machinist and engineer.

On the scientific principle that the adhesion, and consequently the capability of leather belts to transmit power from motors to machines, is in proportion to the pressure of the actual weight of the leather on the surface of the pulley, it is manifest that, as longer belts have more weight than shorter ones, and that broader belts of the same length have more weight than narrower ones, it may be adopted as a rule that the adhesion and capability of belts to transmit power are in the ratio of their relative lengths and breadth. A belt of double the length or breadth of another under the same circumstances, will transmit more than double the power. For this reason it is desirable to use long belts. By doubling the velocity of the same belt, its effectual capability for transmitting power is also doubled.

Good stock is the first requirement of a belt, which, if spongy, will not meet that demand. It must be firm, but pliable; the grain or hair side should be free from wrinkles; the stock should show no irregularities in dressing, but be of an even thickness throughout; the splices should be mathematically true, and if rivets are employed, they should be inserted on the hair side, and the burrs sent home before riveting; the edges should be parallel and perfectly straight. In handling a belt, examine it carefully, double it up, the hair side out, and press it together; if it crack under this treatment, it should be rejected, as the rational use of a belt consists in utilizing the whole amount of power it will transmit.

Belts are sometimes used having a transmitting power of double the capacity necessary where they are employed, while quite as often they are much too narrow for the work required of them. The first instance shows a useless waste of material, the latter poor economy; as, in order that it may perform the work required, it is necessary frequently to take it up, as a result of which the weak points succumb to the strain, and it is torn asunder; or if not, the shaft is likely to be drawn out of line, or the bearing over heated.

In using a new belt a few days, if it present a mottled appearance on the side next to the pulleys, it may be set down that it is not furnishing the full capacity of its power. The spots referred to indicate that certain portions of the belt do not touch the pulley, and that its entire transmitting power is not utilized. If the face of the pulley is true, and the belt is as nearly perfect as possible, the defect may be remedied by the judicious application of rendered tallow and fish oil, two parts of tallow to one of oil, melted and allowed to cool. A new belt should be used a day or two before it is oiled, and frequent application of small quantities are better than too liberal oiling at long intervals.

If a belt, of the proper size for the work it has to do, slip on the pulley, it is caused by the centrifugal force, which tends to throw it outward; a corresponding degree of tension will check the defect.

Belts should be put on by a person acquainted with their use, as the wear of the belt depends considerably on the manner in which it is put on; therefore, the following suggestions, if practised, will be of much service to persons employed in this capacity. The ends to be joined should be cut perfectly square, in order that one side may not be drawn tighter than the other. Good lace-leather, if properly used, will give better satisfaction than any patent fastening.

Where belts run vertically, they should always be drawn moderately tight, or the weight of the belt will not allow it to adhere closely to the lower pulley; but in all other cases they should be slack. In many instances, the tearing out of lace-holes is unjustly attributed to poor belting; when, in reality, the fault lies in having a belt too short, and trying to force it together by lacing, and the more the leather has stretched while being manufactured, the more liable it is to be complained of.

To obtain the greatest amount of power from belts, the pulleys should be covered with leather. This will allow the belts to be run very slack, and give 25 per cent. more wear.

More power can be obtained from using the grain side of a belt

to the pulley than from the flesh side, as the belt adheres more closely to the pulley; but it should be remembered that the belt will not last quite so long, as when the grain, which is very thin, is worn off, the substance of the belt is gone.

Double-leather belts are frequently used; but it is clearly a mistake, as a single-leather one will transmit more of the power than a double one. Double-leather belts run straighter than single ones, as the flank side of one part can be put against the back of the others. A double belt will stand a greater tension than a single one, but a single belt will stand all that should be put upon any belt.

In cases where a belt is incapable of transmitting the required amount of power, and circumstances preclude the possibility of substituting a wider one, the difficulty may be overcome by using two belts of the same width, one on the top of the other. Two belts run in this way will transmit nearly as much power as one belt the width of the two.

How to test the quality of leather for belting.—Cut a small strip of the leather about $\frac{1}{16}$ of an inch in thickness, and place it in strong vinegar. If the leather has been thoroughly tanned, and is of good quality, it will remain for months even immersed, without alteration, simply becoming a little darker in color. But, on the contrary, if not thoroughly tanned, the fibres will quickly swell, and, after a short period, become transformed into a gelatinous mass.

How to make belts run on the centre of pulleys.—It is a common occurrence for belts to run on one side of the pulleys. This arises from one or two causes: 1st, One or both of the pulleys may be conical, and, of course, the belt will run on the higher side. The most effectual remedy for this will be to straighten the face of the pulleys. 2d, The shafts may not be parallel or exactly in line. In this case the belt will incline off to the side where the ends of the shafts come the nearest together. The remedy in this case would be to slack up on the hanger-bolts, and drive the hangers out or in, as the case may be, until both ends of

the shafts become exactly parallel. This can be determined by getting the centres of the shafts at both ends by means of a long lath or light strip of board.

Tighteners.—The tightener should be placed as close to the large or driving-pulley as circumstances will permit, as the loss of power incurred by the use of the tightener is equal to that required to bend the belt and carry the tightening-pulley. Consequently, there is a greater loss of power by placing it near the small pulley, as the belt is required to be bent more than when it is placed near the large one.

The reason why belts run to the highest side of a pulley is due, in part, to centrifugal force, and also to the fact that the part of a belt nearest the highest part of a rounded pulley is more rapidly drawn, because the circumference of the pulley is greater at that point.

Rubber and leather belts.—Rubber belts will transmit nearly as much power as leather belts with the same tension; and they have this advantage, that they may be made of any length, width, or thickness, and yet always run straight, providing the pulleys are in line. Besides, their first cost is much less than those of leather; but they will not last over half as long. They cannot be run in situations where the belt rubs, nor as cross-belts, or through forks, as shifting-belts; and when they give out, it is almost impossible to repair them.

If a rubber belt runs off, and becomes entangled in the machinery, ten chances to one it will be completely ruined; whereas, a leather belt, under like circumstances, will sustain very little injury. When saturated with oil, they soon rot, and when situated in cold, damp places, they are liable to freeze, which has a tendency to separate the different thicknesses and ruin the belt. Besides, they often freeze to the face of pulleys when standing still, and when started up, the gum facing is torn off, which ruins the belt.

A leather belt, if made of good stock, not overstrained and properly treated, will last for twenty years. When partly worn

out, it may be cut up and used over again for a narrower or shorter belt; and when entirely unfit for the transmission of power, it may be used for different purposes around a factory; but when rubber belts are worn out, they are of no value whatever.

To prevent accidents *by shafts revolving within reach of operatives' garments in mills and factories.*—Cover the shaft with a loose sleeve of sheet-tin or zinc, and insert a ring of thick gum or leather at each end, to prevent rattling. Should it become entangled with the garments of any of the operatives, the resistance will cause the sleeve to stand still while the shaft is rotating within it, by which the person may be extricated and accident averted.

Rule *for finding the length of belt wanted.*—Add the diameter of the two pulleys together; divide the sum by 2, and multiply the quotient by 3¼. Add the product to twice the distance between the centres of the shafts, and the sum will be the length required.

Another rule *for finding the length of a belt.*—Add the diameter of the two pulleys together, multiply by 3⅛, divide the product by 2, add the quotient to twice the distance between the centres of the shafts, and you have the length required.

Rule *for finding the width of belt to transmit a given horse-power.*—Multiply 36,000 by the number of horse-power; multiply the speed of the belt in feet per minute by one-half the length in inches of belt in contact with smaller pulley; divide the first product by the second; the quotient will be the required width in inches.

Rule *for calculating the number of horse-powers a belt will transmit, its velocity, and the number of square inches in contact with the smaller pulley being given.*—Divide the number of square inches in contact with the pulley by 2; multiply this quotient by the velocity of the belt in feet per minute, and divide by 36,000. The quotient is the number of horse-powers the belt will transmit.

Another rule.—Divide the number of square inches of belt in contact with the pulley by 2; multiply this quotient by the velocity of the belt in feet per minute; divide this amount by 32,000, and the quotient will be the number of horse-power.

Rule *for finding the change required in the length of a belt when one of the pulleys on which it runs is changed for one of a different size.*—Take three times the difference between the diameters of the pulleys and divide by 2. The result will be the length of belt to cut out or put in.

How to measure a coil of belting.—Add the diameter of the hole, in inches, to the outside diameter of the roll; multiply by the number of coils in the roll; then multiply this by the decimal ·1309, and the product will be the number of feet in the roll. To have the exact length, the average diameter must be used if the roll is not perfectly round, and fractional parts of an inch must not be omitted in the calculation.

How to put on a belt.—Never place a belt on the pulley in motion; always place it first on the loose pulley or the pulley at rest; then run it on the pulley in motion. If the belt is very heavy, and the pulleys run at a very high speed, it is advisable to slack on the speed of the engine; but when this is impracticable or inconvenient, care must be taken to mount the belt on the exact face. The person engaged in so doing must have a firm footing, and prevent his clothing from getting in contact either with the belt or pulley. Where the belt is heavy, and the location such that it is impossible to get a solid footing and exert strength in running on the belt, it is best to stop the engine and mount the belt on the pulley as far as possible. Then take a small rope, double it, slip one end through the arms and around the belt and rim of the pulley, and the other end through the loop formed by the double of the rope; then stand on the floor on the opposite side, and draw on the rope, when the belt will be hugged to the periphery of the pulley. When motion is communicated, it may be slipped on without any trouble, while by letting go the end of the rope when the belt is on the pulley, the noose will be undone and the rope thrown off.

Rule *for finding the required size of a driving-pulley for any required speed.*—Multiply the diameter of the driven pulley by the number of revolutions it should make, and divide the product by

the revolutions of the driver. The quotient will be the required size of driver.

Rule *for finding the diameter of a driven pulley for a given number of revolutions, the diameter and revolutions of the driver being known.*—Multiply the diameter of the driver by its number of revolutions, and divide the product by the number of revolutions of the driven pulley. The quotient will give the proper size of the driven pulley.

Gearing.

Rule *for finding the diameter of toothed wheels.*—Multiply the number of teeth by the number of thirty-seconds of an inch contained in the pitch, the product will be the diameter in inches and hundredths of an inch; or, multiply the number of teeth by the true pitch, and the product by ·3184. These results give only the diameter between the pitch-line, on one side, and the same line on the other side, and not the entire diameter from point to point of teeth on opposite sides. It must also be borne in mind that these results are only approximate diameters, since the wheel often varies from the computed diameter in consequence of shrinkage and other causes.

Rule *for finding the required number of teeth in a pinion to have any given velocity.*—Multiply the velocity or number of revolutions of the driver by its number of teeth or its diameter, and divide the product by the desired number of revolutions of the pinion or driven.

Rule *for finding the diameter of a pinion, when the diameter of the driver and the number of teeth in driver and pinion are given.*—Multiply the diameter of the driver by the number of teeth in the pinion, and divide the product by the number of teeth in the driver, and the quotient will be the diameter of pinion.

Rule *for finding the number of revolutions of a pinion or driven, when the number of revolutions of driver and the diameter or the number of teeth of driver and driven are given.*—Multiply the

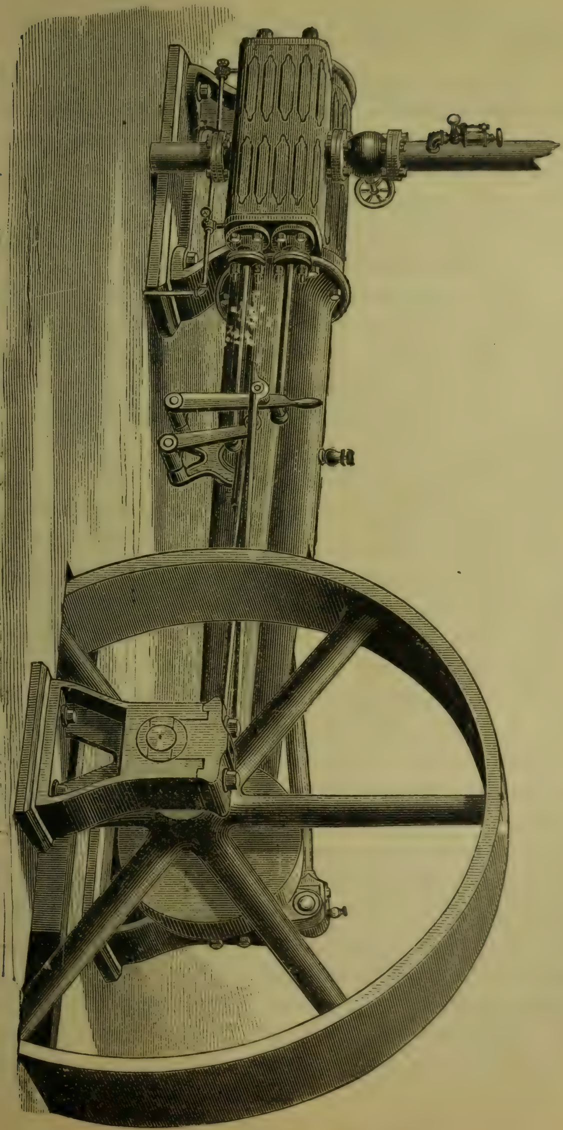
number of revolutions of driver by its number of teeth or its diameter, and divide the product by the number of teeth or the diameter of the driven.

Rule *for finding the number of revolutions of a driver, when the revolutions of driven and teeth, or diameter of driver and driven, are given.*—Multiply the number of teeth or the diameter of driven by its revolutions, and divide the product by the number of teeth or the diameter of driver.

Rule *for finding the number of revolutions of the last wheel at the end of a train of spur-wheels, all of which are in a line, and mesh into one another, when the revolutions of the first wheel, and the number of teeth, or the diameter of the first and last are given.*—Multiply the revolutions of first wheel by its number of teeth or its diameter, and divide the product by the number of teeth or the diameter of the last wheel; the result is its number of revolutions.

Rule *for finding the number of revolutions in each wheel for a train of spur-wheels, each to have a given velocity.*—Multiply the number of revolutions of the driving-wheel by its number of teeth, and divide the product by the number of revolutions each wheel is to make. The result will be the number of teeth required for each.

Rule *for finding the number of revolutions of the last wheel in a train of wheels and pinions, spurs or bevels, when the revolutions of the first or driver, and the diameter, the teeth or the circumference of all the drivers and pinions, are given.*—Multiply the diameter, the circumference, or the number of teeth of all the driving-wheels together, and this continued product by the number of revolutions of the first wheel; and divide this product by the continued product of the diameter, the circumference, or the number of teeth of all the pinions, and the quotient will be the number of revolutions of the last wheel.



Back View of the Fitchburg Automatic Cut-Off Engine.

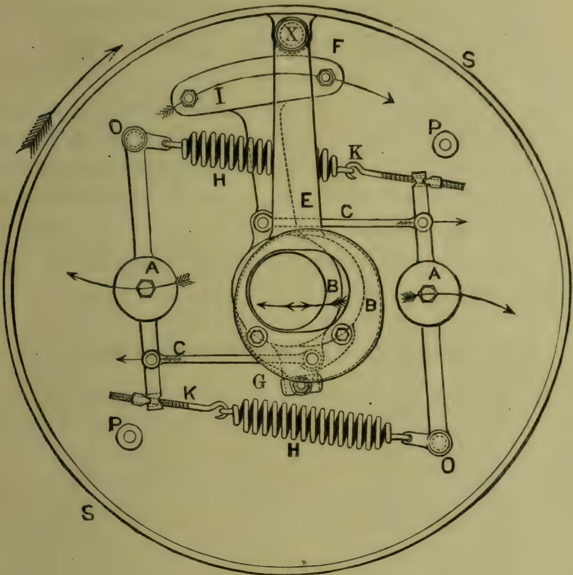
Fitchburg Steam-Engine Company's Automatic Cut-Off Engine.

The cut on page 647 represents the Fitchburg horizontal, automatic, cut-off engine, with positive valve-gear and independent steam and exhaust arrangements. As will be observed, the frame is of the girder pattern, faced up at one end to receive the cylinder and the other the pillow-block. The legs, which support the cylinder and main-bearing, are bolted to foundation-plates, which prevents the possibility of any movement. The steam-valves receive their motion from a movable eccentric, with variable throw, and admits or cuts off the steam at any desired point in the stroke, to meet the requirements of load and pressure. The exhaust-valves are worked by a direct movement from an eccentric keyed on the shaft.

The governor, a cut of which may be seen on page 649, is placed on the main shaft, is enclosed in a disc, and is the same in principle, though differing somewhat in mechanism from that used on the Buckeye engine. It is claimed to be very sensitive and powerful, thus ensuring a steady motion under the most varying loads and steam-pressures, which is, of itself, a desideratum of great importance, as any increase in speed over that at which the engine was intended to run, is a waste of steam, and consequently a waste of fuel; and as any lagging of an engine behind the regular speed at times induces a loss of production. Because, it is well known that to produce economical results, the valve-gear must be so arranged as to admit the necessary volume of steam to the cylinder at the right time, and no more.

S S **is a disc which is firmly keyed to the shaft.** *X* shows the position of the crank-pin in its relation to the other parts; *A A* are weights attached to arms having their fulcra at *O O*; *H H* are coiled springs, attached at one end to the arms at *O O* by means of swivels, and at the other by means of adjustable hooks, *K K*. As will be observed, short stub, connecting-rods, having one end attached to the weighted arms and the other end to the

cast-iron collar, *B*, which is accurately fitted to the shaft, are introduced between the weights and the springs. This collar, *B*, has an arm which extends from one side to the periphery of the disc, and which ends in a flat weight, *G*, to which weights of the



same shape may be attached. On the opposite side an ear, *I*, is shaped to receive a sliding-block; close to the periphery of the disc, *S S*, is pivoted an arm, *E*, its other end encircling the shaft, and which (in consequence of having an oblong slot) admits of a swinging motion across the shaft; at its end opposite the pivot, beyond the part encircling the shaft, is an ear or boss sustaining a steel pin, upon which pin turns the sliding-block which moves in the ear, *I*. Upon the arm, *F*, is bolted the eccentric, which has a sufficient advance to give a ¾ cut-off when the governor is at rest. The spottings, *P P*, are intended for the fulcra

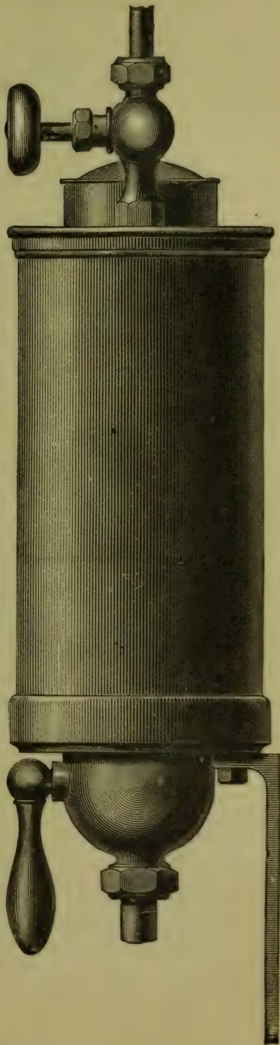
of the weighted arms, in case it should be necessary to run the engine in the opposite direction.

The action of the governor may be explained as follows:—When the engine is travelling below speed, the eccentric is kept in full throw by the tension of the spiral springs, and the steam follows the piston three-fourths of the stroke. As soon as the proper speed is attained, the centrifugal action of the weights, *A A*, overcome the tension of the springs, and they move outwards in the direction of the arrows, thus lengthening the spring. By means of the connecting-rods, *C C*, the outward motion of the weights gives a motion round the shaft to the collar, *B*, which in turn, by means of the ear, *G*, and the sliding-block attached to the arm, *E*, gives the latter an oscillating movement from its point of suspension across the shaft (as shown by the arrows), and at the same time to the eccentric, which is bolted to it.

The Improved Circulating Salinometer.

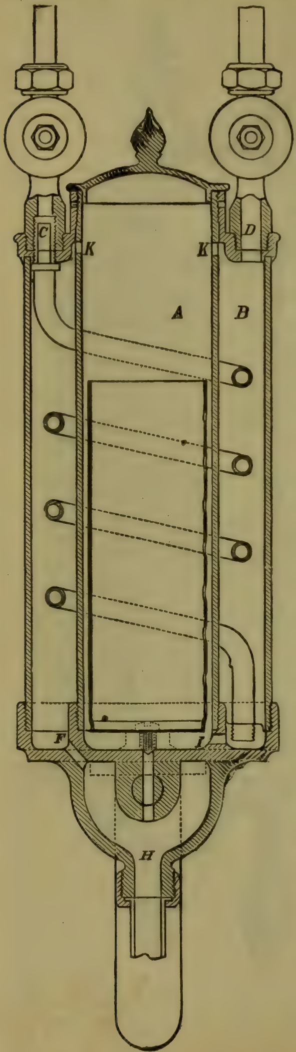
The cut on page 651 represents the Circulating Salinometer, the object of which is to prevent the sputtering or boiling of the water when drawn off from the boiler under pressure, and the consequent inconvenience and danger of scalding. This object is accomplished by reducing the temperature below the boiling-point before it enters the testing-pot. It serves also to keep up a continuous circulation, so that the degree of saturation can be observed at any moment, thus avoiding the necessity of frequently drawing off the water, and saving the time which would be wasted in so doing. This salinometer consists of a large and small pot attached to the same bottom, one inside of the other, with a coil in the annular space between the two pots. This coil is connected at the top with a globe-valve, and at the bottom with a passage leading into the small or testing-cup, which contains a hydrometer and thermometer, the latter being hung by a spring hook on the upper edge of the pot. The right-hand globe-valve is used for admitting cold water into the annular space between

the pots, for the purpose of reducing the temperature of the water in its passage from the boiler to the required degree, so that it may rise in the testing-pot perfectly quiescent, and is kept at the proper height; and the circulation is maintained by openings to the pot near the top, through which it overflows into the annular space between the two pots, after which it escapes with the cold water by the escape-pipe through the passage-way, making but one connection at the bottom. These passages are situated upon each side, right and left, and, should they become choked by sediment resulting from muddy water, they can be cleaned out by disconnecting the escape-pipe and running a wire through them. The plug-cock at the bottom is only used for emptying the testing-pot when required, and has no communication with anything else.



The water from the boiler and the cold water can in no way become mixed except by the bursting of the coil, which is not likely to happen unless it should be left full of water and allowed to freeze. The coils are thoroughly tested before they are put in; the top is not specially designed to be tight, as nothing can escape from the joints but the cold water, and that only when the annular space is allowed to become full, which is unnecessary, as a very small quantity of cold water

will be sufficient to reduce the temperature of the water passing



through the coil. A hole is drilled in the back of the large pot, near the top, which allows the water to escape in case it should accidentally become full. The cold water is supplied by a pipe connected by a globe-valve to any pipe or valve below the water-line supplying cold water, and led to the salinometer. If it should be desired to place the salinometer above the outside water-level, the cold water can be supplied by some of the pumps.

In erecting these salinometers, they may be secured to the boilers or bulkhead, but when there are two or more boilers, a very neat and convenient arrangement may be made by fitting them close together on a plain cast-iron plate fastened down with tap-bolts, and with the pipe for the cold water fitted just above them, with a **T** coupling and branch to each one, the plate being secured with tap-bolts in any convenient place in the engine-room. A salinometer may be attached to each boiler, and all of them supplied with cold water from the same pipe, or one may be connected with two or more boilers.

It is preferable to have one connected with each boiler, as in that case the density of the water may be observed in any boiler independent of the others. To put the salinometer in

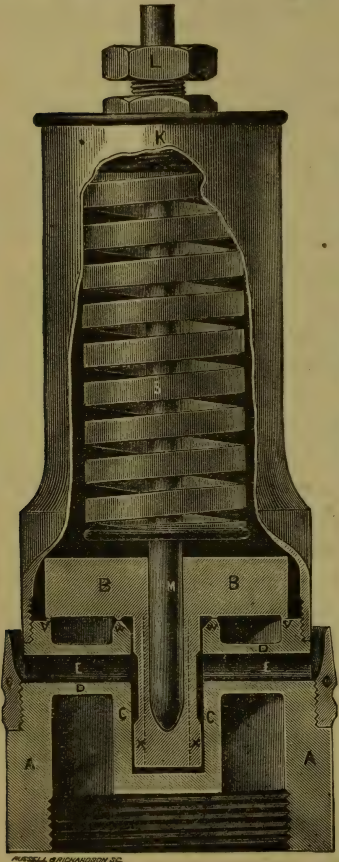
operation, it is only necessary to open the valve communicating between the boiler and the salinometer, and admit the water as fast as the overflow will allow it to escape, and when the temperature reaches the required degree, the communication may be sufficiently closed only to allow the necessary circulation. Then the cold-water valve may be opened, and, if the connection with the boiler has considerable length of pipe to retain cold water, the blow-off cock may be opened to admit of its escape, and then closed. A very slight opening of either valve will be sufficient to keep up the circulation and keep the water at the required temperature. If the water should be admitted too rapidly from under pressure, the agitation of the water at the bottom, even below the boiling-point, will disturb the hydrometer. When the circulation and temperature are properly adjusted, it will not require to be touched from one end of a passage to another, unless it may be to adjust the cold-water valve occasionally if any considerable change takes place in the temperature of the water in the boiler.

The manipulation and operation of these salinometers are very simple and satisfactory, as they are a decided improvement on any other arrangement of the kind ever heretofore used for the same purpose, as, with one of them attached to each boiler, the density of the water may be accurately determined at any moment, which is a feature of great importance in many respects, and a fact which will be appreciated by those who have used other arrangements. The different parts of the salinometer are designated as follows: *A*, hot-water pot; *K*, outlet for hot-water overflow; *B*, cold-water reservoir; *H*, general outlet; *C*, hot-water inlet to coil; *e*, outlet passage from hot-water pot; *D*, cold-water inlet; *f*, small pot for hydrometer; *I*, passage from coil to hot-water pot; *F*, outlet for cold water.

These salinometers are manufactured by the Crosby Steam Gauge and Valve Company, Boston, Mass.

Crosby's Adjustable "Pop" Safety-Valve.

The annexed cut represents Crosby's Adjustable "Pop" Safety-Valve.—Its mechanism may be explained as follows: The valve proper, *B B*, rests upon two flat annular seats, *V V* and *W W*, on the same plane, and is held down against the pressure of steam by the steel spiral spring, *S*. The tension of this spring is caused by screwing down the threaded bolt, *L*, at the top of the cylinder, *K*. The area contained between the seats, *W* and *V*, is what the steam-pressure acts upon, ordinarily, to overcome the resistance of the spring. The area contained within the smaller seat, *W W*, is not acted upon at all until the valve opens. The large seat, *V V*, is formed on the upper edge of the shell or body of the valve, *A A*. The small seat, *W W*, is formed on the upper edge of a cylindrical chamber or well, *C C*, which is situated in the centre of the shell or body of the valve, and is held in its place by four arms, *D D*, radiating horizontally at right angles to each other, and connecting it with the body or shell of the valve. These arms are hollow and form four passages, *E E*, for



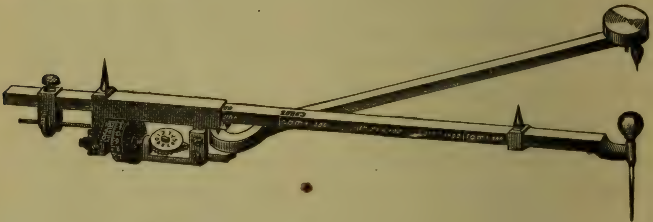
the escape of the steam or other fluid from the well into the air when the valve is open. This well is deepened, so as to allow the

wings, *X X*, of the valve proper to project down into it far enough to act as guides. The area of the apertures at the outer ends of the passages through the arms is reduced more or less at will by screwing up or down the adjustable ring, *G G*.

Action of the "Pop" Safety-Valve when under Pressure.—When the pressure under the valve is within about one pound of the maximum pressure required, the valve will open slightly, and the steam will escape under the larger seat into the cylinder surrounding the spring, and thence into the air. The steam is also forced under the smaller seat into the well, and thence, through the passages in the arms, into the air. As soon as the pressure attains the exact maximum point, the valve will be lifted so high as to force the steam into the well faster than it can escape through the apertures in the arms. A pressure will then accumulate under the inner seat, which will be in excess of what was required to overcome the increasing resistance offered by the spring, and, acting upon the additional area presented, at once forces the valve wide open, and rapidly relieves the boiler. This pressure under the inner seat is of itself differential. The valve then at once slowly settles down, and the pressure under the inner seat as slowly diminishes. This action continues until the area of the opening under the smaller or inner seat is less than the area of the apertures in the arms for the escape of the steam; the pressure then ceases and the valve promptly closes. The point of opening can be readily changed while under steam by screwing the threaded bolt at the top of the cylinder either up or down, and the point of closing is as easily adjusted by screwing up or down the ring surrounding the outside body or shell of the valve.

This valve is automatic, certain in its action, prompt in opening and closing at the required points of pressure, and can be fully relied upon to relieve the boiler under all circumstances. Experience and use have confirmed the following claims for it, namely, opens precisely at fixed working pressure; discharges all excess of steam above fixed working pressure; reduces the pressure rapidly upon opening; closes with the least possible loss of steam;

the limits of pressure within which the valve will open and close are adjustable; uniform in action at different pressures; simple in arrangement, and easily connected and adjusted; does not deteriorate under continued use; never sticks on seat; makes comparatively little noise in discharging; occupies less room than any safety-valve. These valves are made to correspond with the requirements of, and are used on, locomotive, portable, steamboat, stationary, and steam fire-engine boilers, and for other purposes. Each of these valves is tested under steam pressure, and set to open at the exact point of pressure desired, and is adjusted to close at about two pounds reduction. Both of these points may be readily changed by the operator without removing the valve from the boiler or reducing steam. Any person of ordinary intelligence will readily understand the principle and operation of these valves.



The Improved Planimeter.

The above cut represents the improved Planimeter as especially adapted for ascertaining, from the indicator diagram, the average pressure in the steam-engine cylinder, and also for measuring the superficial contents of regular or irregular plain surfaces. It is claimed to have the advantages over any other in use, in being supplied with a supplementary wheel, with a graduated plate, marked with figures representing ten times the value of the figure on the roller-wheel, thus saving the care and trouble incidental to the use of the other single-wheel instruments, and in giving the average height of the indicator diagram in one-fortieth of

an inch (instead of the area), which, multiplied by a factor representing the scale or number of the spring used, gives the average pressure in pounds, without the long process and trouble of measuring the length of the diagram, dividing it into the area, and then multiplying by the vertical scale.

It is also adapted for measuring the superficial contents of regular or irregular plain surfaces, and representing the contents either in millimeters, inches, feet, perches, or acres, as the operator may desire, by adjusting the sliding-bar. In the case of indicator diagrams, if the Crosby Indicator be used, the process of finding the area of the diagram is simplified, as the springs used are of such scales (mostly multiples of four) that, instead of the long process formerly used, the mean pressure is obtained by simply multiplying by a factor corresponding to the scale used, as follows:

Spring............	8	12	16	20	24	30	32	40	48
Factor............	0·2	0·3	0·4	0·5	0·6	0·75	0·8	1·0	1·4

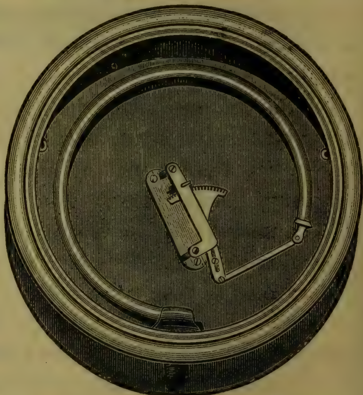
The numbers engraved upon the sliding-bar, *A*, serve for the calculation of the contents of surfaces, for which special instructions are required. The arms of these planimeters are made hollow and composed of the best grade of German silver, the whole instrument being made with great precision, accuracy, and skill. They are manufactured by the Crosby Steam Gauge and Valve Company, Boston, Mass. The same firm makes two other styles of planimeters, one corresponding with the common instrument in use, which has only one wheel, as shown on page 323, and another similar to it, having two wheels.

Crosby's Improved Steam-Pressure, Hydraulic, Combination, Vacuum, and Self-Testing Gauges.

Steam Gauges.—About the year 1849, Eugene Bourdon, of France, discovered that the free end or ends of a flattened metallic tube possessed of sufficient elasticity for use as a spring, would move when pressure was exerted through the medium of a

fluid applied externally or internally; that the motion was in direct proportion to the pressure applied; and that when the pressure was removed they would assume their former position. From this circumstance, he conceived the idea of a new pressure gauge, in which the bent tube should be the main spring or means of motion. But, though it was generally conceded at that time that the hollow tube spring gauge, as invented by Bourdon, excelled in delicacy and sensitiveness any previous mechanical arrangement employed for that purpose, nevertheless, it was demonstrated by experience that such a device, owing to its peculiar construction, was not well adapted for all the purposes for which pressure gauges are employed, as, in consequence of being held only at one end, it would vibrate from a sudden shock or slight change of pressure, thus causing the pointer to oscillate on the dial-plate, inducing friction and wear, and rendering the indications of the gauge uncertain and delusive. Besides, the dip of such a spring caused it to retain a portion of the water condensed in it, thus rendering it liable to burst in cold weather, to be strained by freezing, and lose its tension.

Exterior View of Crosby's Steam Gauge.


Interior View of the Original Bourdon Steam Gauge.

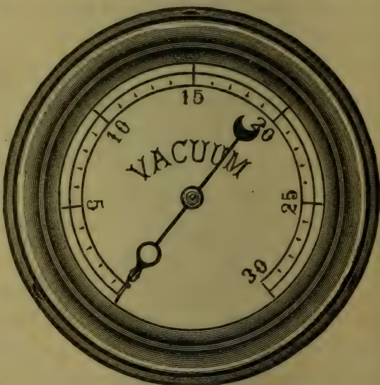
To overcome these defects, numerous devices have been suggested and tried, but they almost invariably embodied the same defects as those above mentioned, and were subject to the same errors, the gravest of which arose from the straightening or setting of the springs. Steam users are more indebted to George H. Crosby for remedying the foregoing defects in pressure gauges, and for the production of a perfectly reliable steam gauge, than to any one previous to his time, as he discovered, by observation and experiment, that only the horizontal motion of the free ends of the springs or tubes, while under varying pressure, had been used heretofore, and that they had a perpendicular or upward action, as well, when the springs were of proper length and shape, and that by uniting these motions by proper mechanism, it could all be transmitted to the pointer. In accomplishing this, he discovered that a firmer and stiffer spring than any heretofore used for the same pressure was an absolute necessity. And as a result, no pressure over that indicated by the pointer on the dial will affect their original elasticity, and vibration of the pointer under varying pressures is obviated; besides, in consequence of the spring being held at the lowest points, they have no dip, which arrangement admits of the water returning to the siphon, thus preventing freezing. Thus it would seem that, while the Crosby gauges embrace all the desirable points in the original Bourdon gauge, they also embody many others which have been demonstrated by experience to be absolute necessities in the construction of an accurate, reliable, and serviceable steam gauge.

Interior View of Crosby's Steam Gauge.

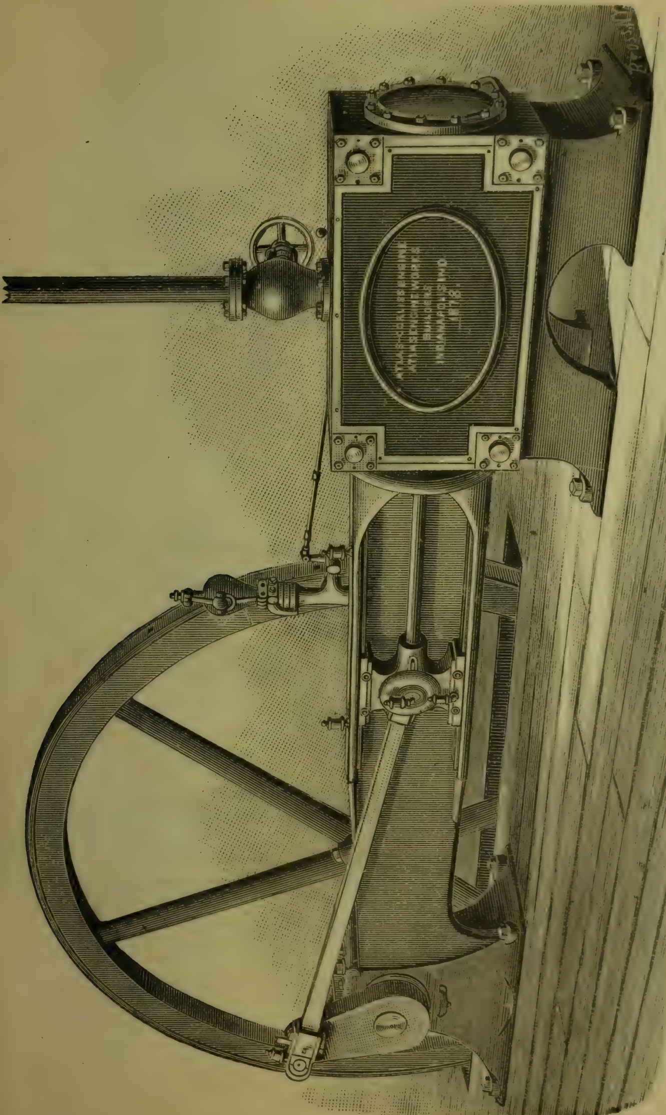
Self-Testing Steam Gauges.—This class of gauges is of great importance, convenience, and utility, as the engineer in charge can always ascertain whether his gauge is correct or not by observing the following instructions: Set off all pressure that may be on the gauge, after which the pointer will fall to zero; then unscrew the plug on the left-hand side, which uncovers the hook. To this hook hang the first weight by the spindle. This is marked by a certain number, and the pointer should travel at once to the corresponding number on the dial, if correct at this point. But if the pointer stands below or above this number, it will indicate just how much the gauge is "out," and in which direction. Proceed by adding the next higher numbered weight, and continue as before.

Crosby's Self-Testing Steam Gauge.

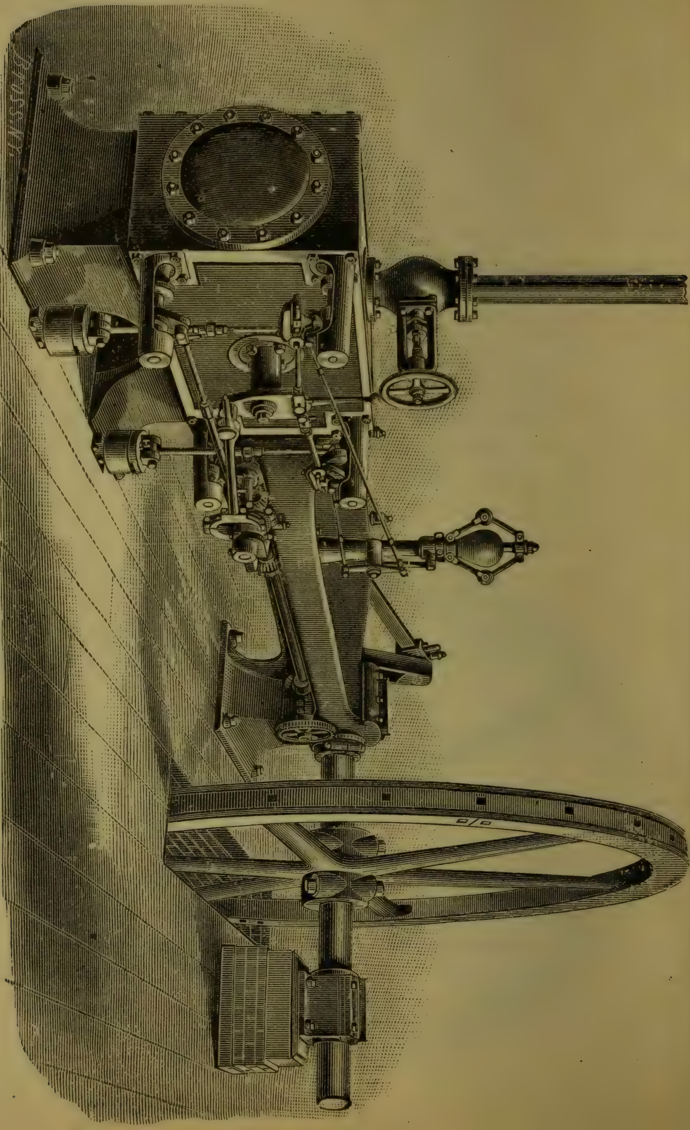
Vacuum Gauges.—The conditions under which vacuum gauges act are the reverse of steam gauges, as, in the vacuum gauge, the interior of the tube is influenced by the vacuum, while its exterior is exposed to the action of the atmosphere. These gauges are manufactured by the Crosby Steam Gauge and Valve Company, Boston, Mass., and are all tested by a mercury column before being put in use.



Crosby's Vacuum Gauge.



Front View of the Atlas Corliss Engine.



Back View of the Atlas Corliss Engine.

The Atlas Corliss Engine.

The cuts on pages 661, 662, show a front and back view of the Atlas Corliss engine. As will be observed, the frame is of the girder pattern; a form which has been more extensively copied, for the past twenty years, by engineers and steam-engine builders both in this country and Europe, than any other. Though a Corliss engine in every respect, it differs from others of the same type in many very important features; one of which is, that the main frame, hind leg, and main-bearing are cast in one solid piece, which is not generally the case with other Corliss engines, as, in most instances, the main-bearing and its supports are cast separately, and bolted to the frame; another is, that the horizontal section of the frame is deeper and heavier than in most Corliss engines, while a deep rib, running to the base of the legs, insures additional rigidity and stiffness. The frame, as in the case of all engines of the Corliss type, is faced up at the front end, to receive the cylinder, which rests on a pedestal of ample proportions. In consequence of the large metal surface brought in contact with the foundation, the weight of the engine is more uniformly distributed, and the jar, which is so detrimental to the stability of many types of engines, is entirely obviated.

While the steam- and exhaust-valves and the cut-off arrangements are essentially the same as in most Corliss engines, the mechanism which works the valves and controls the cut-off, is entirely different. In the ordinary Corliss engine, only one eccentric is employed to operate the steam- and exhaust-valves, through the medium of a wrist-plate, which must be so connected with the eccentric, as to change the direction of its motion at the proper time for opening and closing the steam- and exhaust-valves. To accomplish this object, the eccentric must be placed nearly at a right angle with the crank; in consequence of which its direction changes at about half-stroke; the result of which is, that the cut-off is limited to the preceding portion of the stroke, as the clutch must be detached during the forward motion. In

the Atlas engine, this difficulty is remedied, as two eccentrics are used — one for the steam- and the other for the exhaust-valves, each of which is set independently, for the most accurate performance of its own work. The exhaust eccentric has nearly the same angular position as the single eccentric in ordinary Corliss engines. The cut-off eccentric is placed nearly 90 degrees behind it, and therefore does not change the direction of the cut-off clutch, until a correspondingly later period in the stroke is reached, which is a very important feature in itself.

The manner in which the eccentrics receive their motion is different from that generally employed, as, instead of being rotated on the crank-shaft, they are placed on a supplementary or counter shaft, which has the same motion as the main shaft, and is situated directly under the cylinder end of the frame. This eccentric shaft is operated through the medium of gears from the main shaft, through a side shaft, which is located directly under the horizontal rib of the frame. The side shaft also operates the governor, thus dispensing with the governor-belt and its necessary risk and uncertainty. The governor is of the "Porter" type, which has been successfully applied to engines on which most other governors have failed to give satisfactory results. This is due to the fact, that the heavy centre weight gives the constant force of gravity acting downwards; while the centrifugal force of the rapidly revolving balls is the variable force, and acts upwards through the joints of the governor; the result of which is, that the governor rises or falls, as the variable force is greater or less than the constant force. It is very powerful and sensitive, and holds the engine in perfect control under the most varying circumstances of load and pressure. It is also provided with an automatic stop, which becomes operative in case of accident.

The valves and cut-off mechanism are essentially the same in the Atlas as in most other Corliss engines, as may be seen in the cuts on pages 284, 285.

A is the valve-stem, as shown in Fig. 1. *B* is a bell-crank fastened to the valve-stem, by which motion is communicated to the

valve. *C* is the cut-off clutch, which is made of gun-metal and faced with hardened steel. *D* is a case-hardened block, having a large bearing in the bell-crank, *B*, which allows it to adapt itself freely to any angular position. This block is virtually a part of the bell-crank, and contains a hole at right angles to the axis of its bearing, through which the small end of the rod, *F*, which

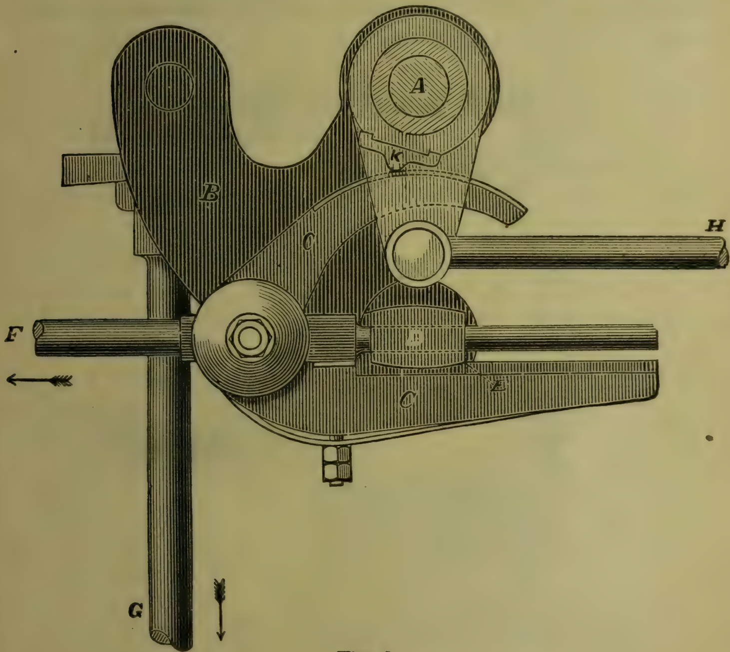


Fig. 1.

carries the cut-off clutch, is passed, and receives its motion from the cut-off eccentric by means of a rocking-plate on the side of the cylinder. *G* is the dash-pot rod, to which a weight is attached, for the purpose of closing the valves promptly when the cut-off is effected. *H* is the governor-rod; it varies the angular position of the governor-toe, *K*, as the governor rises and falls, and so

determines the time of cut-off. *K* is the governor-toe, and is supported on a bushing concentric with the valve-stem. The cut-off occurs when the governor-toe, *K*, depresses the cut-off clutch, *C*, sufficiently to detach it at the point, *E*, from the bell-crank, *B*, allowing the unsupported weight of the dash-pot to close the valve by its fall.

A cross-section of the cylinder through the steam- and exhaust-ports is shown in Fig. 2.

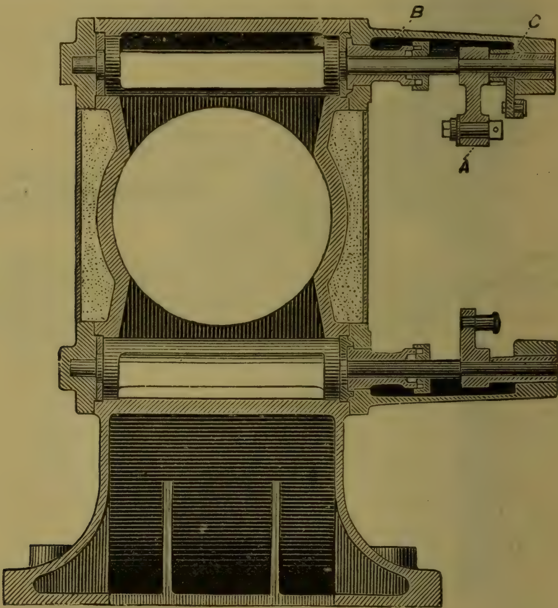


Fig. 2.

A **is the bell-crank,** connected by the cut-off clutch directly to the rocking-plate. *B* is the stuffing-box, which has a very long bearing between the ground-joints of the collar and the gland at the outside. *C* is an out-board bearing fitted with a bushing, the inside of which forms a bearing, the cut-off toe of the governor

being carried on its outer surface. The advantage of this arrangement is, that it prevents the valve-stems from springing, which would have a tendency to increase their friction, and cause them to wear out of round; while, in consequence of the brackets being hollow, they form a receptacle for the drips from the valve-stem, stuffing-boxes, and insure perfect drainage. There are many points of *excellence* to be noticed in the design and construction of these engines, among which are simplicity of design, convenient arrangements for accurate adjustment of the different parts, and independent steam- and exhaust-valve motions. The cross-head bearings are flat, and so arranged that they may be repaired or renewed at short notice and trifling expense. Besides, the cross-head wrist-pin is placed both in the horizontal and vertical centre lines of the bearing surfaces, thus relieving the cross-head of the excessive weight and severe strain incident to an overhanging connection.

The Atlas Corliss engines are built of excellent material, thoroughly fitted, and tastefully finished. The bearings for the rubbing, reciprocating, and revolving surfaces are ample, thus preventing the possibility of rapid wear and the necessity of expensive repairs. The fly-wheels are turned on the face and sides, and accurately balanced, which insures smooth running; while the cylinders are covered with "asbestos" and cast-iron lagging, which prevents condensation and insures economy. The steam-piston packing used is Babbit & Harris's patent (an illustration of which may be seen on page 167), which has been generally adopted by the builders of the best class of steam-engines in the country, and has the reputation of giving entire satisfaction.

The Atlas engines are built, both condensing and non-condensing, of any power to meet the requirements of purchasers, and for whatever purpose employed, whether for milling, manufacturing, or pumping, have the reputation of giving entire satisfaction. They are manufactured at the Atlas Corliss Engine Works, Indianapolis, Indiana.

Questions,

THE ANSWERS TO WHICH WILL BE FOUND IN THE TEXT.

What is acceleration?

Define the term affinity.

What constitutes an angle?

Explain the principle embraced in the use of axles.

Give the meaning of the term attraction.

What is meant by capillary attractions?

Define the terms gravity and centre of gravity.

Give the meaning of the terms adhesion and cohesion.

Under what two heads may elastic fluids be classified?

Define the term elasticity.

Has the term energy any definite meaning when applied to mechanics?

What is force?

Define the term focus as used in geometry.

What is meant by the term friction?

Give the meaning of the terms hydrodynamics, hydrostatics, and hydraulics.

Explain the formation of the hyperbola.

Define the term impact.

What is meant by the term impenetrability?

Define the term impetus, as applied to mechanics.

What is meant by the incidence, as applied to mechanics?

Explain the meaning of the term inclination.

To what class of the mechanical powers does the inclined plane belong?

What is the meaning of the term inertia?

Under what three classes may levers be divided?

Give the definition of the term machine.

What physical elements are embraced under the head of mechanics?

What is meant by the modulus of the elasticity of any substance?

Give the meaning of the term momentum.

What is meant by motion, as applied to mechanics?

Enumerate the different kinds of motions.

Define the centre of oscillation.

Explain the mechanical principles represented in the vibration of the pendulum.

Define the centre of percussion.

Why is perpetual motion an impossibility?

Explain the meaning of the term pneumatics.

Of what two forces is power the product?

What is the difference between pressure and weight?

What kind of machines may be termed prime movers?

What is the mechanical principle involved in the use of the pulley?

To which of the mechanical powers is the screw most nearly allied?

Give the meaning of the term resilience.

Define the science of statics.

What is meant by strength, when applied to mechanics?

Enumerate the different kinds of strength.

Give the meaning of the word tools.

Define the term torsion.

What is meant by the term velocity?

What is understood by the term weight?

Under which of the mechanical powers may the wheel and axle be classed?

What mechanical principles are embodied in the wedge?

Give the atomic weights and chemical equivalents of the different metals in use at the present day.

Which is the most useful metal?

Give the weight of a cubic foot of wrought- or cast-iron.

Give the component parts of cast-iron.

Of what two elements is steel composed?

Give the heat-conducting properties of copper, brass, cast- and wrought-iron.

Give the tensile strength of copper, brass, gun-metal, wrought- and cast-iron.

Give the proportions of carbon in the various grades of iron and steel.

Give the different proportions of the metals which form the basis of brass, Muntz metal, gun-metal, Babbitt metal, and bronze alloy.

Give the composition of fusible metal which melts at a temperature of 212° Fah.

Give the rule for finding the approximate weight of iron castings from the weight of the pattern.

Give the shrinkage various metals undergo in the process of casting.

Give the weight and bulk of several substances in cubic feet, pounds, and tons.

Give the extension of wrought- and cast-iron at various temperatures.

Does wrought-iron increase in heat up to a certain temperature?

Is the tensile strength of copper increased by the application of heat?

Give the composition of a good non-conductor for preventing radiation of heat in steam-boilers, cylinders, pipes, etc.

Give the component parts of a good durable cement for steam-joints.

What advantages does belting possess over cog-gearing, and *vice versâ*?

From what principle do belts derive their power to transmit motion?

What are the necessary characteristics of good belting?

What conditions are necessary to obtain the greatest percentage of power from any belt?

What advantages have double and single belts, and *vice versâ*?

How would you test the quality of belting?

How would you proceed to make belts run on the centre of pulleys?

What are the advantages of rubber belts over leather, and *vice versâ*?

Give the rule for finding the length of a belt.

Give the rule for finding the number of horse-power a belt can transmit.

Give the rule for finding the change in length in a belt when one of the pulleys is changed.

Explain the most practicable method of putting on belts.

State the precautions to prevent accidents when throwing on or taking off belts.

Give the rules for finding the size of pulleys for any required speed.

INDEX.

THE END.